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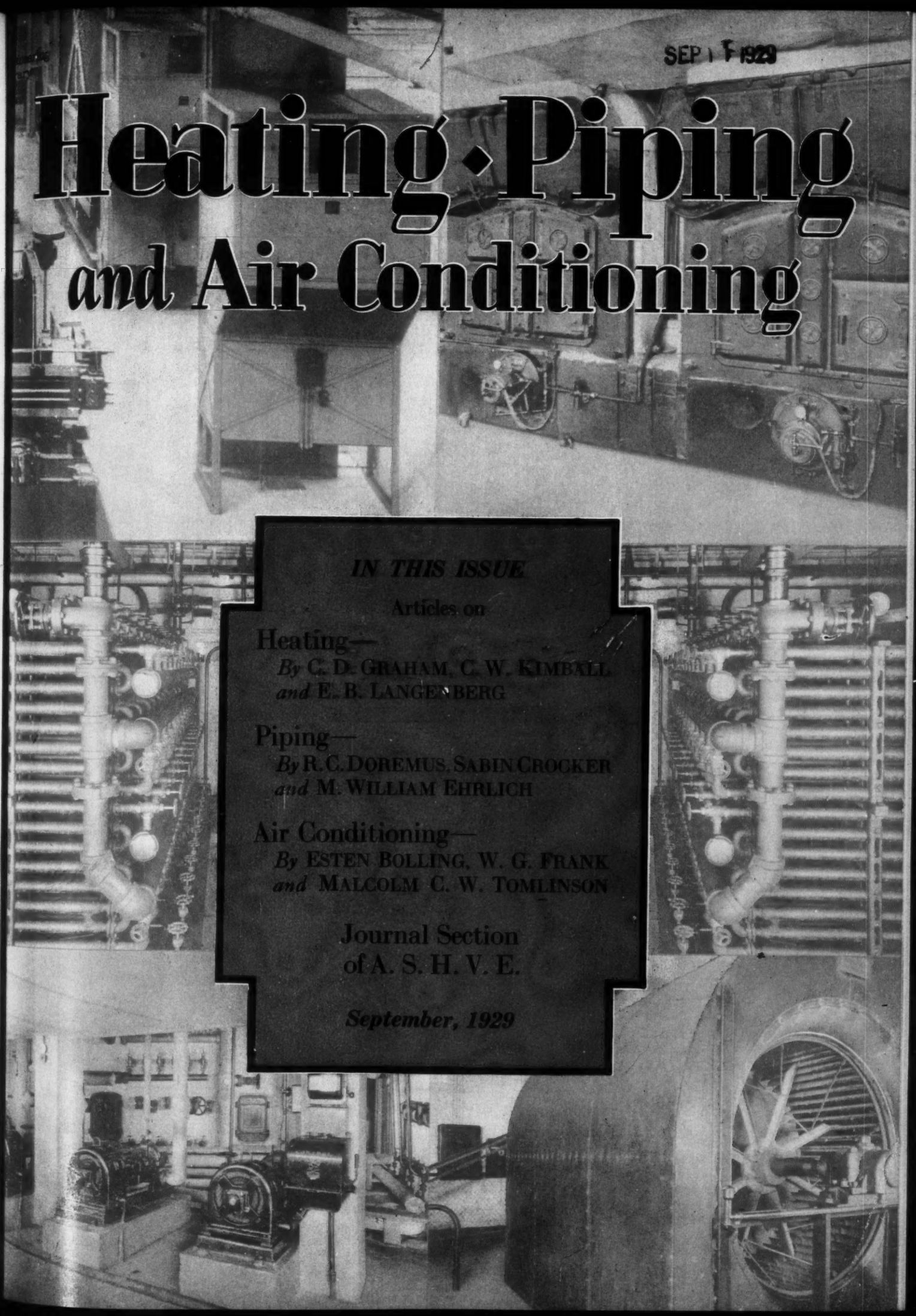
Heating · Piping and Air Conditioning



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Air Conditioning—
By ESTEN BOLLING, W. G. FRANK *and* MALCOLM C. W. TOMLINSON

Journal Section
of A. S. H. V. E.

September, 1929

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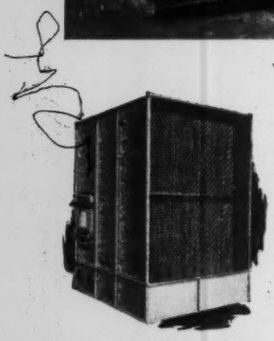

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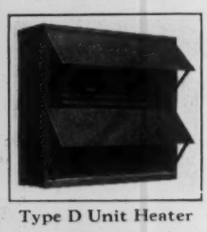

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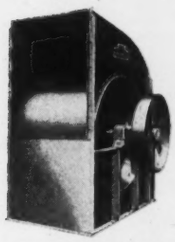
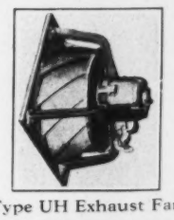


HV Ventilating Fan



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The EDITOR'S PAGE

Heating · Piping and Air Conditioning

PRESENTING A NEW DEPARTMENT

There are some new "faces" this month—new, that is, to our columns but not new to the field in which they hold high positions. Before introducing them, however, we want to introduce a new department—where our readers become writers. "Open for Discussion" is the name of this department, and in it we follow the practice of technical societies of giving space in their convention programs for discussion of the papers presented. Articles presented in this publication are practically the same as papers presented at technical society meetings. The writer is not necessarily the only engineer who has something to say on the particular subject. Therefore, other engineers who want to add to it or ask questions about it to bring out more information can consider it "open for discussion." It should become an interesting department.

OUR PLAN FOR A BALANCED EDITORIAL CONTENT WORKING OUT NICELY

We have mentioned before on this page the manner in which we aim to cover our field so that there will be in each issue a proper proportion of heating articles, a proper proportion of piping articles and a proper proportion of air conditioning articles. That involves more than so many heating articles, so many piping articles and so many air conditioning articles. It involves the kind of article in each class so that, over a period of time, everything pertaining to the heating of large and industrial types of construction will be touched upon, everything pertaining to the piping of steam, high and low pressure, of water, air, oil, gas, refrigeration, hydraulic and process liquids, everything pertaining to air conditioning both in its application to manufacturing processes and in its application to human comfort and health.

It is a big order, but it is the one we set out to fill when we published our first issue five months ago. Each issue has built upon the last, getting more and more into the field which one issue alone could hardly cover. The five together, which will include this one, add up to 450 pages of technical material—an evidence of the amount of ground we have already covered The big point, though, is that it has been proportioned—has been comprehensive.

It is material that has been prepared by some of the best engineers in the world. We have appreciated the co-operation of these engineers which has made it possible for us to offer such valuable data and information. It is not one-time co-operation, either. These engineers are maintaining their interest in our plans and contributing regularly so as to insure our ability to continue month in and month out on the high standards we have set.

A QUARTET OF NEW AUTHORS APPEARS IN THIS ISSUE

We want to express considerable satisfaction in the opportunity we have of presenting Esten Bolling, consulting engineer, as a contributor. Mr. Bolling is recognized as one of the leading air conditioning engineers in the world, and he combines an engineering knowledge with an unusual gift of writing. His article, "Fundamentals of Air Conditioning," which not only starts out this issue but also starts out a series of several that he will write for us, makes air conditioning thoroughly clear in its basic principles. And, packed with real information to do this, it is written in a style that will be quickly appreciated. An article as long as this is frowned upon by some publishers. We believe our readers will agree that the manner in which Mr. Bolling handles it makes length an advantage rather than an objection.

Another new contributor this month is R. C. Doremus, of George B. Bright Co., refrigerating engineers and architects in Detroit, Mich. The type of work Mr. Doremus' company is constantly doing involves a thorough knowledge of refrigeration piping design, installation and maintenance and makes him, therefore, an engineer whose writings on "Refrigeration Piping," the title of his article, carry the weight of experience.

C. D. Graham is chief engineer of the York Heating & Ventilating Co., Philadelphia. "Making the Unit Heater Fit the Job" is his article, and Mr. Graham has had to select the right unit heater for the conditions encountered so often that he is certainly qualified to cover the subject.

M. William Ehrlich has contributed a chart from which pipe sizes, steam flow and pressure drop for lines carrying saturated and superheated steam can be quickly and easily computed. Mr. Ehrlich is an engineer whose work in designing piping systems has been simplified by the chart he has worked out. The work of other piping engineers can be equally simplified. It is a chart which can be cut out and made part of the designer's "tools."

Pipe Joint Design

Sabin Crocker's series on the design of flanged pipe joints continues in this issue. Mr. Crocker has compiled some absolutely new data in the material we have the privilege of submitting, and his articles on this subject should become a part of the designer's engineering library.

Ideas and Personalities

Another new feature appears in this issue — "Interviews of Interest." In the conversations of people who have something to do with heating, piping or air conditioning there crop up many worthwhile ideas and suggestions. In themselves they are not adaptable to a complete article. But they mean something to the engineer and contractor, and should be passed on to our readers. You will find such things in "Interviews of Interest," and if they really are of interest they will become a regular feature.

Warm Air Data

Warm air heating takes prominence again in this issue among our heating articles. Warm air is adaptable to the heating of many of the larger and industrial types of construction, and the consulting engineer, architect and contractor should know its possibilities. E. B. Langenberg presents a "data article" this month, one which tells how to heat a church with warm air.

It is our aim to present data on every type of heating that is adaptable to the larger and industrial types of construction.

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Published Monthly by Engineering Publications, Inc. 1900 Prairie Avenue, Chicago

TERMS OF SUBSCRIPTION
Anywhere in the world, $3.00 per year.
Current copies, 50c each. Back numbers, $1.00 each.

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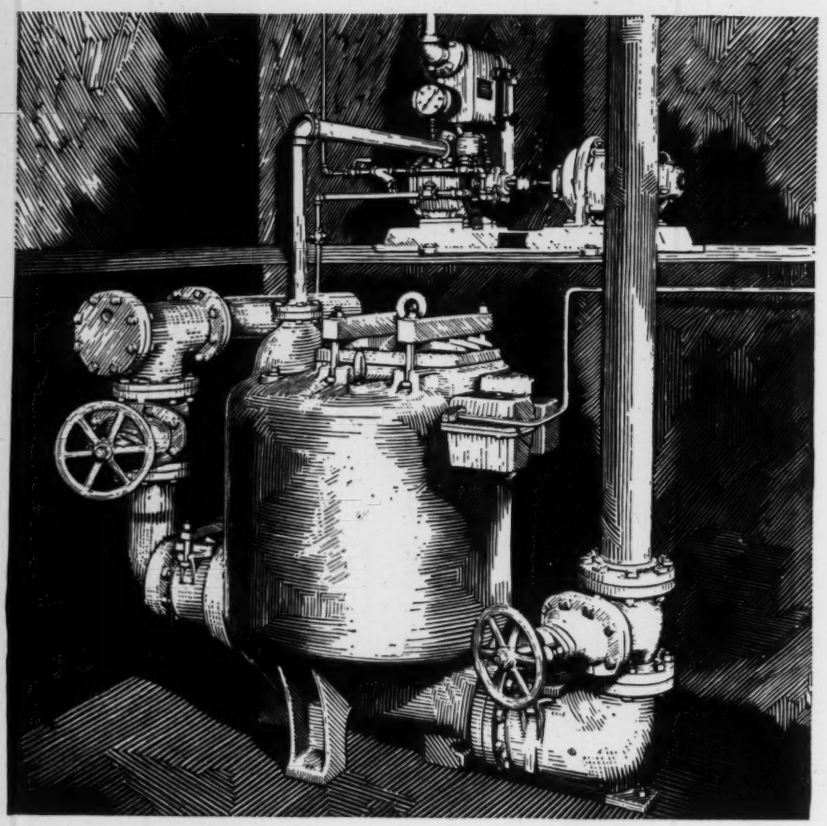

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Heating · Piping
and Air Conditioning

Vol. I | September, 1929 | No. 5

The Fundamentals *of* Air Conditioning

CENTRAL STATION AIR CONDITIONING SYSTEM IN A TEXTILE PLANT

By Esten Bolling, Consulting Engineer

Who Says:

It is hoped that those not familiar, in detail, with the various principles and practices of modern air conditioning will find in this writing a brief but comprehensive treatise which will stimulate as well as guide their further interest in this rapidly expanding, increasingly important art.

AIR conditioning is so modern an art that its early history may be dismissed with but a brief review of its high points. In its true sense air conditioning had its inception about 1901, when Willis Carrier, then an employee of the Buffalo Forge Company, began to study the possibilities of treating air mechanically, preparing it for specific purposes and then distributing it effectively that it might accomplish those purposes, the whole system being controlled automatically to insure uniformity and dependability. Since 1901, not spectacularly, but slowly, laboriously, painstakingly, air conditioning has been developed into a great science contributing mightily,—in many cases so marvelously that its accomplishments

strain belief—to the practices and profits of, not ten, but hundreds of basic industries, and to the comfort and health of countless thousands of human beings.

It seems irrelevant to detail that early history, except to state boldly that to the knowledge of heating; to the knowledge, after a fashion, of distributing; and to the knowledge of "washing" air were added, first, means of humidifying it; then, mechanical control of those means; and, later, means of dehumidifying and cooling, with similar automatic control. Cleaning, from the first, was a concomitant effect of the water-spray used for humidification or dehumidification and cooling, though the cleaning or "washing" was sufficiently thorough to meet most requirements.

As development progressed the apparatus and its auxiliaries were improved and air distribution, a vital factor in air conditioning, was studied intensively until wholly adequate and remarkably adaptable systems were perfected.

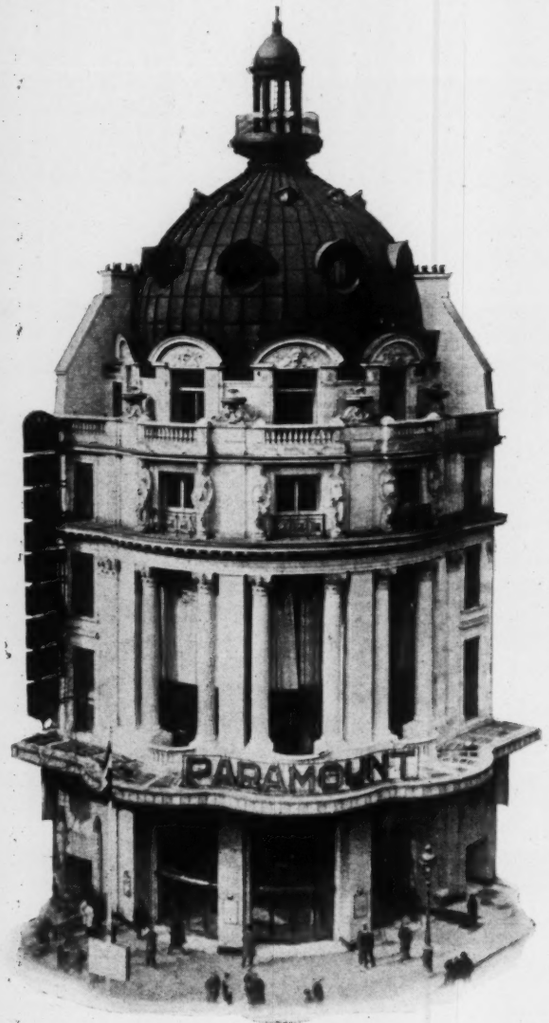
Air conditioning must not be confused with ventilation, or heating, or cooling, or humidification or dehumidification, or filtration. It is none of these. It is all of these plus a scientific coordination which requires consummate engineering with highly developed skill in design, installation and adjustment to the given service.

Like all tremendously potential sciences, air conditioning has been hampered by the practices of those unqualified either technically or temperamentally, or both.

In no other really scientific development have there been so many opportunities for charlatanism. This is true because the ultimate purchaser of air conditioning equipment has been, until recently, wholly ignorant of the results to be expected or demanded, and even now the most atrocious engineering may go entirely unnoticed because the equipment accomplishes so much, —so miraculously much,—that its detailed faults are overlooked by the owner ignorant of such details.

Every fan engineer knows that a simple ventilating system can be greatly reduced in initial cost by the use of skimped duct work and too small a fan. The purchaser pays a lesser price for the installation and then for the rest of its shortened life pays a heavy premium upon its operation,—perhaps without ever knowing it.

In air conditioning it used to be as easy to "get away" with such practice, for the purpose of "getting the job," as it still is in ventilation, but more recently,—because of the vaster volumes of air handled, the higher costs of conditioning it as compared to removing it alone,—the tremendous value of experienced engineering has become apparent even to the owner who knows or cares little about the technical details of the installation. Today the owner will readily pay a few dollars more for a fan selected to operate at the peak of its carefully-determined performance curve when the modern engineer-salesman (note the precedence) points out to him that the reduction in motor horsepower at so much per Kw./Hr. will pay not 6 per cent but, mayhap, 80 per cent upon his small differential investment.



PARAMOUNT THEATRE, PARIS, FRANCE—A THEATRE COMPLETELY AIR CONDITIONED FOR HEALTH AND COMFORT

And today the owner is learning that the surest way to lose money is to try to save it in the initial cost of air conditioning equipment. For two reasons—(1) the heavy operating expense of badly designed apparatus, (2) the inadequacy of such an installation, which may cause the total loss of his investment or so seriously impair the result-value that the installation becomes worth only a small part of its "lower" price.

As the owner has learned so has the general public. The still prevalent sign upon the ornate facades of sundry movie theatres, that it is "Twenty Degrees Cooler Inside," fools fewer and fewer people every year. The man-in-the-street cannot long be trifled with. He has learned that a mere sign, despite its dangling painted icicles, and a great rush of air through the entrance doors, do not connote air conditioning, with its attendant comfort and health.

Mr. and Mrs. Average Person—and don't you forget, Miss and Master Average Person also — are rapidly learning the vast difference between a real air conditioning system, with adequate refrigeration for summer cooling and dehumidification, and anything less. They are learning this through conditioned movie theatres, of which there are now hundreds here and many abroad, department stores, restaurants, hotels, office buildings, great indoor stadiums like that of "Paddy" Harmon in Chicago. Soon they will learn it through other applications of air conditioning which will directly and inescapably affect the public at large, and about which we may have something to say here in future numbers.

This general appreciation of air conditioning on the

part of the public has resulted and, in the near future will result more noticeably, in increasing industrial applications. Dad will hear from son or daughter, or observe for himself, at the movies, in the store or club, that air conditioning, expertly applied, simply eliminates weather troubles and he will take (he is taking) the natural mental step, "Why not use this in my business?"

Perhaps he's known for years that Jim or John, engaged in a similar business, had some sort of dingfangled equipment that made him more or less independent of outdoor weather. But now he realizes that such equipment is an everyday fact, that it positively does make the indoors wholly independent of the outdoors, that it is widely and successfully used for all manner of purposes, in practically every climate with which this terrestrial sphere is afflicted or blessed, as the viewpoint may be.

Then will air conditioning come into its own. The day is near.

Until 1923, when the first scientifically designed air conditioning system was placed in operation in a motion picture theatre, there had been but a handful of applications contemplating, primarily, the comfort and health of human beings. Prior to this the major development of air conditioning had been industrial. This is not surprising when it is realized that the industrial applications permitted at least partial evaluation of results in cold dollars and cents. In many cases it was possible to show that the air conditioning system, by eliminating certain manufacturing difficulties due to the hygroscopicity of the materials involved, had paid for itself during the first month or two of its operation! Not only this, but the accomplishments of air conditioning in increasing production, with the former floor-space, mechanical equipment and personnel are not short of marvelous.

Air conditioning completely revolutionized the textile industries—cotton, wool, silk and rayon. In fact, the foremost manufacturers of rayon, that magic textile of the chemist, freely admit that without scientific air conditioning of the highest order their remarkable commercial development would have been impossible.

Not only has air conditioning played an indispensable rôle in the processes of textile manufacture, but it has pushed back the climatic frontiers once supposed to circumscribe, for instance, the locales of cotton looming. The great textile industry of Lancashire grew up because the fortuitous circumstance of the district's awful climate (extremely moist) is advantageous to cotton manufacture, though unhealthy and depressing to the worker.

A similar textile center evolved in our own New England, because of a similar, though not so awful, climate.

For generations Alabama cotton was shipped to England or New England for fabrication. Then air conditioning altered the whole industry. Cotton mills, equipped with scientific air conditioning, sprang up in the midst of cotton fields. The cotton industry had gone to Dixie, where the cotton blossoms grow—because air conditioning had removed climatic considerations. As a matter of fact, it must be pointed out, ere progressing, that air conditioning had been adopted by most of the New England mills and many of the foreign mills years before the exodus—because there's no such thing as "favorable" outdoor weather, which can, at best, be but occasionally favorable, whereas air conditioning is always favorable, now, then, here, there. Instead of dependence upon the proverbial vagary of "the weather," air conditioning means the independence of mechanical weather, as you like it, when you want it.

There is a vast and growing cotton manufacture in India, made possible by the experienced ministrations of British air conditioning engineers trained right here in the United States, the country in which the prophets of air conditioning have been, indeed, without honor—until very recently. Just recently in "Time," the indispensable and usually extraordinary well-informed, we read a story of Lancashire and cotton and air conditioning, though that proper title was not used. They say "the thing (air conditioning) has been done in Germany." To be sure it has—but it has been a primary economic factor of first importance here for two decades, while American engineering and American equipment are so far the finest in the world that American installations have been and are being installed not only in Germany but in practically every other civilized country.

We should take a great national pride in our development of air conditioning—the modern worker of miracles, the guardian of health, the assurance of comfort and efficiency. It behooves us, then, as engineers and Americans, to know our air conditioning.

We should take a great national pride in our development of air conditioning—the modern worker of miracles in industry, the guardian of the human being's health, the assurance of his comfort and efficiency.

The quantity production of fine confections, at reasonable costs and independent of season, would have been impossible without air conditioning. Bread, and many other food products, rubber tires, aeroplanes, automobiles, chemicals of many sorts, film, film studios and laboratories, clay products, lacquers, leather and its products, such as shoes, paper, printing and lithography, soap, tobacco in all its forms, machine tools, pistols and rifles, gunpowder, razor blades, pianos, phonographs, radios, broadcasting studios, clocks, electric lamps, glass, celluloid, flour and so on—more than 200 distinctly different industries—have been benefited signally by modern air conditioning. This is an amazing record. It ranks air conditioning among the major inventions of our time.

It behooves us, then, as engineers and as Americans, to know our air conditioning!

Not in the restricted sense, not by considering controversial details or specific applications, but broadly, understandingly, that we may assist its development and protect it from both misapprehension and misapplication.

To read, as I frequently have, that air conditioning is "growing up" (editorial comment in this very magazine —*touche!*) or that it is "advancing gradually," or had its inception when someone built a fire back in the dark ages, for purposes of keeping warm, does, I must confess, amuse me.

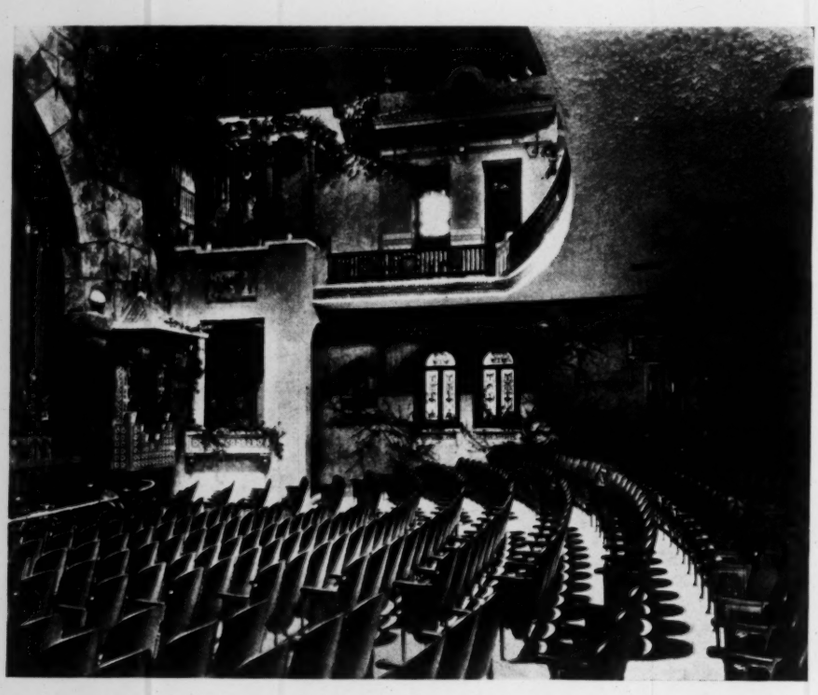
Air conditioning is already grown to full stature as a vitally important asset of modern life. It is not advancing gradually. It is now advanced to an extraordinarily

high plane of engineering development. It did not begin ages ago, any more than steam or internal combustion transportation began when man first rode a dinosaur. It began when it was conceived that neither heating, nor cooling, nor ventilation, nor cleansing, nor humidification, nor dehumidification, was, in itself, sufficient, but that all of these, simultaneously and technically controlled, were essential to the proper utilization of that circumambient element without which the human being succumbs in a few moments and the world would disintegrate.

Because the effects of impure or improperly conditioned air upon the human being (short of its complete absence) are insidious, whereas impure water or other forms of faulty sanitation are more or less immediately catastrophic, sanitary scientists long neglected air. Then, as our urban congestion increased, as we began to build subways and tunnels, and we discovered, of necessity, the process or industrial rôles of the air, we began to realize its vital part in the comfort and health of the human being—apart from its primary function as the source of oxygen.

This writer delights in pointing out that air is without question, above all other things, prerequisite to human life. We can do without food for weeks, without water for days, but without air we yield the ghost in a matter of seconds.

Inanimately air is equally important, even when its oxygen is not required.



Auditorium of El Encantro Theatre, Havana, Cuba—the First Theatre in a Tropical Climate to Be Air Conditioned for the Health and Comfort of Patrons

Because, as water is the universal solvent, air is the universal carrier—of heat, of moisture, of oxygen and ozone, of nitrogen, of many other gases, the undesirable ones of which it renders less harmful by dilution and dispersion. Unlike water, air, compressible, is tremendously useful in countless ways. Air can be made a useful, efficient servant or it can become an insidious, pitiless enemy. Air conditioning is the science which effectively utilizes and controls the manifold functions of air. It is already indispensable in countless ways and it will become more and more so as we continue to develop our less and less natural, more and more artificial modes of living.

Within but a brief span of less than three decades air conditioning has grown to almost unbelievable importance, industrially and humanly. Tomorrow, as such things go, air conditioning will be a commonplace necessity in every enclosure wherein are human beings or their wares, in process or finished. Today real air conditioning—not a makeshift—is available for the home—for winter service, in most cases, but for summer service in a few instances, the latter equipment being well on the way to development.

That we may share effectively in the great development of air conditioning which is so rapidly progressing, let us review the fundamentals upon which further progress depends.

Air Conditioning

Everybody knows that air is a mixture of gases, chiefly oxygen and nitrogen, about in the proportions of one to four. Air nearly always contains certain impurities, the proportion being much greater in congested centers, and CO_2, being a product of exhalation and of complete combustion, is so universally present (about in the same proportions everywhere except when artificially concentrated by some local condition) that it may be regarded as a normal constituent of the air.

Air as a Carrier

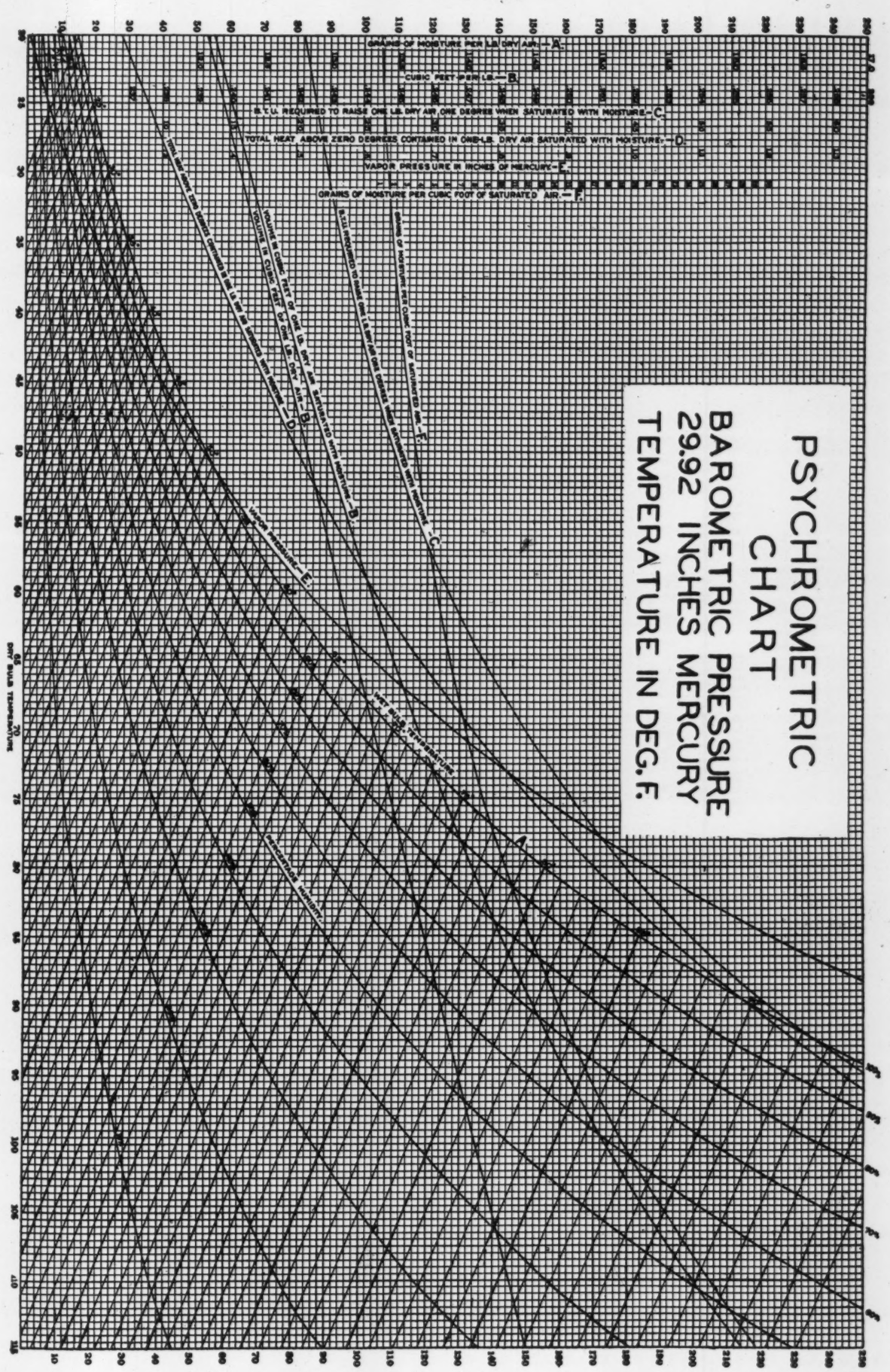
The most vivid conception of air's physical functions is revealed, to our mind, when air is regarded as a carrier or conveyor upon which may be placed heat or moisture for conveyance to or from an enclosure which it is desired to condition. Air may, of course, be also employed to carry other gases or certain solids. Unless carefully cleansed it will invariably carry foreign matter which often is unhealthful or deleterious. It sometimes carries the active aerobic organisms of mold or decay or disease. Of late it has become important that a column of moving air will carry, as well as cause, a noise, oftimes inaudible but disastrous when amplified through a microphone. (Broadcasting. "Talkies.")

The Properities of Air

1—HEAT

The temperature indicated by an ordinary thermometer is the dry bulb temperature. The scale of an ordinary thermometer, either Fahrenheit or Centigrade, is simply an arbitrary standard by means of which comparisons can be established.

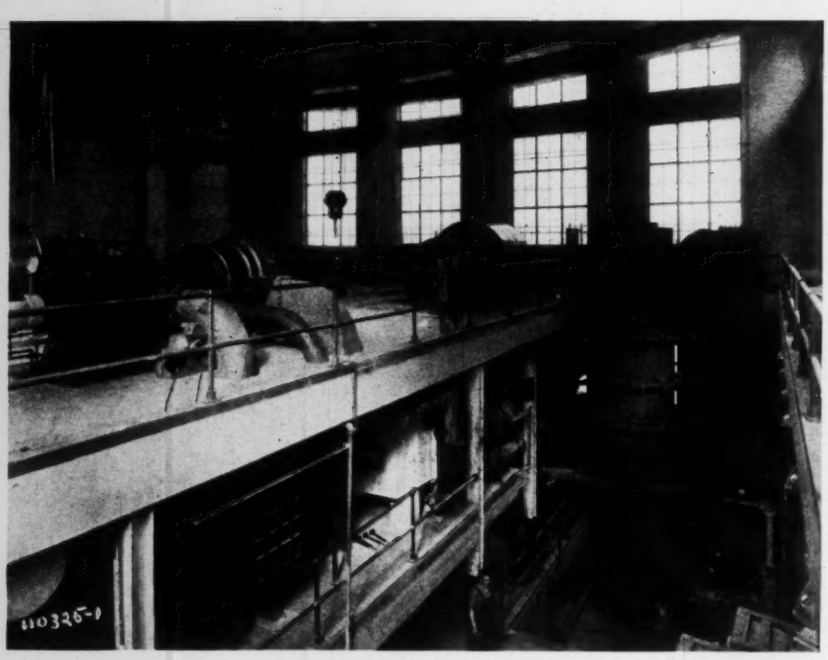
Without involving a too-technical discussion of heat


PSYCHROMETRIC CHART
BAROMETRIC PRESSURE 29.92 INCHES MERCURY
TEMPERATURE IN DEG. F.
GRAINS OF MOISTURE PER LB. DRY AIR.—A.
CUBIC FEET PER LB.—B.
B.T.U. REQUIRED TO RAISE ONE LB. DRY AIR, ONE DEGREE WHEN SATURATED WITH MOISTURE.—C.
TOTAL HEAT ABOVE ZERO DEGREES CONTAINED IN ONE LB. DRY AIR SATURATED WITH MOISTURE.—D.
VAPOR PRESSURE IN INCHES OF MERCURY.—E.
GRAINS OF MOISTURE PER CUBIC FOOT OF SATURATED AIR.—F.
VOLUME IN CUBIC FEET OF ONE LB. DRY AIR SATURATED WITH MOISTURE.—B.
B.T.U. REQUIRED TO RAISE ONE LB. DRY AIR ONE DEGREE WHEN SATURATED WITH MOISTURE.—C.
GRAINS OF MOISTURE PER CUBIC FOOT OF SATURATED AIR.—F.
VAPOR PRESSURE.—E.
DRY BULB TEMPERATURE
WET BULB TEMPERATURE
PERCENTAGE HUMIDITY
A

as molecular motion, let us consider heat from the standpoint of its effects—upon human beings and upon materials.

Heat flows from one position to another, according to its degree of temperature, just as water flows from one position to another, according to its degree of pressure. The dry bulb temperature measures the degree, or intensity, of the heat, or, it may be said, the dry bulb temperature measures the degree of pressure which causes the heat to move from one position to another. Let us for a moment consider a non-hygroscopic material which neither absorbs nor gives off moisture and hence is not affected by the humidity of the air.

Such a material at a temperature of 60 deg. fahr., if surrounded by air at 90 deg. fahr., will take up heat from the air until both have reached the same temperature. At the beginning of the heat transfer, the rate of flow of heat from the air to the material will be greater than at any other time, because the temperature difference or temperature head (90—60=30 deg.) is greater at that time, constantly decreasing as the air and the material approach the same temperature. The rate of heat transfer is thus proportional to the temperature difference between the two.



THE SPECIALLY CONSTRUCTED TOWER TYPE UNITS FOR THE DEHUMIDIFICATION OF COMPRESSED AIR IN THE PLANT OF THE GILLETTE SAFETY RAZOR CO., BOSTON, MASS. THE CENTRIFUGAL REFRIGERATION MACHINE USED FOR COOLING THE WATER SUPPLIED TO THESE UNITS IS SHOWN ON THE LEFT, WITH THE STEAM-DRIVEN COMPRESSOR ON THE UPPER BALCONY, THE SHELL AND TUBE COOLER AND CONDENSER ON THE LOWER BALCONY

Thus, if it is desired to heat a certain material as quickly as possible, the dry bulb temperature of the air surrounding the material must be raised as many degrees as practicable above the desired temperature of the material, so that the rate of heat flow from the air to the material will be as rapid as possible.

Pursuing the conveyor example, the speed of the conveyor represents the temperature difference between the air and the material being heated. Thus, a great temperature difference will effect a rapid transfer of heat.

In the case of hygroscopic materials, however, the dry bulb temperature does not indicate the heating capacity of the air, nor does it, even, indicate the rate at which the heating effect of the air will be consummated. This is true because hygroscopic materials are affected as much by the moisture present in the air as by the heat.

Thus, an hygroscopic material may absorb moisture so rapidly that, due to the latent heat so regained, it may become hotter than the dry bulb temperature of the air itself, and actually give off heat to the air.

Later in this writing the phenomena of latent heat are more fully discussed. It is the intention here to point out that, in the case of hygroscopic materials, which are those with which air conditioning is principally concerned, the dry bulb temperature alone is of relatively small importance, and that the factor of moisture must be carefully considered.

2—HUMIDITY

Air is capable of holding, as a mechanical mixture with itself, varying quantities of water-vapor, depending upon its temperature. When there is mixed with the air all of the water-vapor which it (or more accurately the space which it occupies*) can hold, the air is said to be saturated.

Air (or the space it occupies) which is not fully saturated will readily take up water-vapor when in the presence of water, the water, as it is evaporated, being converted from a liquid into a vapor. On the other hand, if partly saturated air is reduced in temperature, until the amount of moisture present corresponds to the amount which the air is capable of holding at the given temperature, it will, obviously, become saturated air. If the temperature of the air is further reduced, its ability to hold moisture being reduced accordingly, the excess moisture will be condensed, which means that it will be converted from a vapor into a liquid, the reverse of the process which occurred as the air absorbed the moisture.

The process of converting liquid water into water-vapor requires a great quantity of heat. This heat is used in performing the conversion (disgregation work) merely, the temperature of the liquid and the vapor being the same at the end of the process.

If the conversion is from liquid to vapor, we speak of the Latent Heat of Evaporation; if from vapor to liquid, of the Latent Heat of Condensation.

Thus, when air absorbs moisture, that is, when it is humidified, the latent heat of evaporation must be sup-

* Author's Note—Technically, air does not "absorb" water-vapor, though this is the common expression. The vapor, of course, occupies the space with the air, and the same quantity of vapor would be present, under given conditions, were the air present or absent. It is simpler and more direct (and not misleading) to use the familiar conception that the "air absorbs or gives up" the water-vapor and this has been done herein.

plied either from the air or otherwise. And conversely, when the moisture from the air is condensed, the latent heat of condensation, equivalent to the latent heat of evaporation, is recovered.

Cold air is saturated when it contains very small quantities of water-vapor, whereas warm air is not saturated until it contains much larger quantities of vapor.

For instance, air at zero is saturated when it contains but one-half of one grain* of water-vapor per cubic foot. Air at 70 deg. is saturated when it contains 8 grains of vapor per cubic foot, while at 83 deg. 12 grains per cubic foot are required to saturate.

Absolute Humidity

The quantity of water-vapor actually present in a given volume of air, without regard to its temperature, is called its absolute humidity. Thus, if air, at any temperature, contains 4 grains of water-vapor per cubic foot, its absolute humidity is expressed as 4 grains per cubic foot.

Relative Humidity

Since air may contain varying quantities of water-vapor at the same temperature, it is necessary to express its degree of saturation as related to its temperature. Thus, there has been established the term relative humidity. Relative humidity is an expression, in percentage, of the degree of saturation of air at any given temperature. A relative humidity of 50 per cent means that the air, at its given temperature, contains 50 per cent of the water-vapor required to saturate it at that temperature.

For example, 70 deg. air is saturated when it contains 8 grains of vapor per cubic foot. Assume that it contains 4 grains per cubic foot. Its absolute humidity is 4 grains per cubic foot. Its relative humidity is 50 per cent since 4 is 50 per cent of 8.

Similarly, if the same air were heated to 83 deg., its absolute humidity would still be 4 grains per cubic foot, but its relative humidity would be but 33 1/3 per cent, because 12 grains of vapor per cubic foot is required to saturate air at 83 deg.

$$\frac{4 \times 100}{12} = 33\ 1/3 \text{ per cent.}$$

Obviously, therefore, the relative humidity depends equally upon both the factors of temperature and moisture content A variation in either alters the relative humidity.

A clear conception of the relation between temperature and moisture content is essential to an appreciation of air conditioning, and the effects of air upon the materials it surrounds, because it is the relative humidity (not the absolute humidity) which determines the effects of air upon such materials.

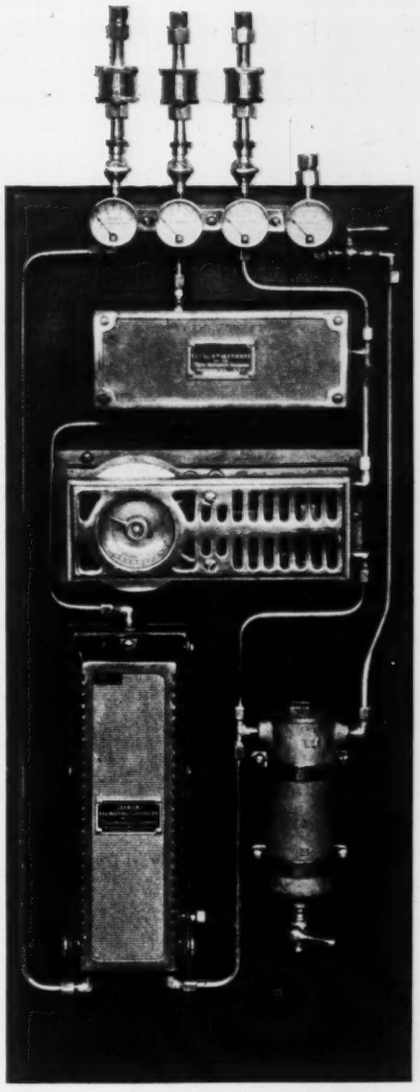
The relative humidity is the governing factor because the drying or moistening effect of air depends upon the relation between the vapor pressure of the moisture in the material exposed to the air and the vapor pressure of the moisture mixed with the air itsef. The vapor pressure of the moisture mixed with the air is proportional to the relative humidity of the air. Since this relation is true, calculations are usually based upon relative humidity rather than relative vapor pressure, although the latter might be considered the more direct relation. It is unnecessary, in a writing of this nature, to inject the more complicated discussion which vapor pressure would involve, and we will therefore continue to employ the more understandable relation of relative humidity.

* 7000 grains = 1 lb.

The Moistening Effect of Air Upon Materials

Air at high relative humidiities (regardless of the absolute humidity) exerts a greater moistening effect than air at lower relative humidities.

For example, air at 83 deg. is saturated when it contains 12 grains of water-vapor per cubic foot. Assume that it contains 8 grains per cubic foot. Its absolute humidity is, then, 8 grains per cubic foot; its relative humidity 66 2/3 per cent.



A Temperature and Humidity Control Panel as Used in Industrial Installations, Such as Textile Mills

Such air will exert a greater moistening effect than air at 92 deg. containing the same 8 grains of vapor per cubic foot. This is true because the 92 deg. air requires 16 grains per cubic foot to saturate it and its relative humidity is therefore but 50 per cent. Thus, although the absolute humidity, or the actual vapor content, of the 83 deg. air is the same as that of the 92 deg. air, the 92 deg. air exerts less moistening effect upon the materials it surrounds because its relative humidity is lower. To express it tersely:

The moistening effect of the air varies approximately with its relative humidity, without regard to the actual weight of water-vapor present.

In a textile mill, for instance, where one of the chief functions of air conditioning is to control the moisture in the yarns in course of manufacture, it should now be obvious that temperature control is equally as important as moisture control, since it is upon the relative humidity (water-vapor content as related to temperature), that

the moistening effect of the air depends.

The Drying Effect of Air Upon Materials

Conversely to the proposition set forth above, 83 deg. air containing 8 grains of moisture per cubic foot, its relative humidity being 66 2/3 per cent, exerts a greater drying effect than the same air at 70 deg., containing the same 8 grains of vapor per cubic foot. This is entirely obvious because the 70 deg. air, containing 8 grains of vapor per cubic foot would be saturated (relative humidity 100 per cent), and would, therefore, exert no drying effect whatever.

Expressing it briefly:

The drying effect of air varies approximately inversely with its relative humidity—the greater the relative humidity, the lesser the drying effect.

But note especially that it is the relative humidity which determines the effect, and that, therefore, the effect depends upon both the temperature and the water-vapor content of the air—since relative humidity depends upon both these factors.

The Heating Effect of Air

The quantity of heat which dry air contains is very small, because its specific heat is low, 0.2415 for ordinary purposes,* which means that one pound of air falling through one degree of temperature (fahr.) will yield but 0.2415 of the heat which would be available from one pound of water, reduced 1 deg. in temperature. The presence of water-vapor in the air materially increases the total heating-capacity of the air because of the latent heat of the vapor itself.

Latent heat is not frequently available by condensation, except in drying or processing, because the operating conditions are usually above the plane of condensation, but all hygroscopic materials are constantly absorbing moisture when in the presence of water-vapor, and as this moisture is absorbed, it is condensed, the latent heat of condensation being yielded to the absorbing material, thus heating it.

As a matter of fact, most hygroscopic materials, in the presence of dry air, even at high dry bulb temperatures, may actually be cooled rather than heated. This occurs because the dry air immediately begins to evaporate moisture from the material, and in so doing removes from the material, as well as from the air, the latent heat of evaporation.

Thus the conditioning of hygroscopic materials depends as much upon the moisture in the surrounding air as upon the dry bulb temperature of the air.

Hence, the conditioning of hygroscopic materials necessitates the control of the moisture content of the surrounding air, as well as its dry bulb temperature.

The Wet Bulb Temperature

There is a second temperature of the air which measures its degree of moistness just as the dry bulb temperature measures its degree of heat.

This second temperature is called the wet bulb temperature.

The wet bulb temperature can be read from an ordinary thermometer by covering the bulb with a moistened wick, and causing the air to pass rapidly over the wick, either by means of a small fan, by aspiration, or by rapidly whirling the thermometer.

The wet bulb temperature is the temperature of evaporation, or, better, the temperature at which the air would become saturated if moisture were added to it without the addition or subtraction of heat.

The wet bulb temperature, in conjunction with the dry bulb temperature, is an exact measure of the heat content of the air, and each must be known.

One of the Air Conditioned Departments in the Jackson Mills, Nashua, N. H.

Knowing the dry and wet bulb temperatures, the absolute humidity, the relative humidity and the total heat of air can be readily ascertained by reference to a psychrometric chart.

Thus, air having a dry bulb temperature of 70 deg., and a wet bulb temperature of 61 deg. is known to contain 4.71 grains of water-vapor per cubic foot, to have a relative humidity of 59 per cent, and to possess a total heat of 26.7 B.t.u. per pound. At the same dry bulb temperature, but a wet bulb temperature of 55 deg., the air contains but 2.87 grains of moisture per cubic foot, has a relative humidity of but 36 per cent, and a total heat of only 23.0 B.t.u. per pound.

From this example, it is again apparent that the dry bulb temperatures and the wet bulb temperature must both he controlled if the effects of air are to be regulated.

Which is but another way of saying that both the temperature and the moisture content must be controlled coincidentally, if air conditioning equipment is to be effective.

* Willis H. Carrier, Rational Psychrometric Formulae.

The Dew-Point

In order to explain the methods of control utilized in modern air conditioning, it is necessary to discuss, briefly, a third temperature of the air, called the dew-point temperature. This is the temperature of saturation, or the temperature at which a given body of air would become saturated if it were cooled to that temperature without the addition or subtraction of moisture.

The temperature of saturation is called the dew-point, because it is the temperature at which the moisture in the air begins to condense, in the form of tiny droplets, or dew.

Methods of Control

There are several methods employed for the automatic control of temperature and humidity, an important one of which is that known as the dew-point control.

Air of extremely low relative humidity exerts a tremendous drying effect upon human beings and materials. In the case of humans, this effect is extremely injurious, being the chief cause of our winter ailments—coughs, colds, grippe, depression, influenza, lassitude.

Dew-Point Humidification Control

With this control, the dew-point, or saturation-temperature, of the air is automatically controlled by means of a simple expansion thermostat, exposed to the air at the instant of saturation in the air conditioning machine itself. Thus, the absolute humidity of the air is definitely fixed in the conditioning machine, because, as has been pointed out, where air is saturated at a given temperature, it contains a given quantity of water-vapor corresponding to that temperature. Obviously, any absolute humidity (i. e., any given number of grains of water-vapor per cubic foot of air) can be established by adjusting the thermostat to the corresponding temperature.

The saturated air leaving the machine is heated by passage through suitable heaters, and its dry bulb temperature is increased sufficiently to establish the required dry bulb temperature in the space being conditioned. Obviously the dry bulb temperature of the air entering a space to be heated must be higher than the temperature to be maintained in that space, so that upon diffusion, or admixture, the desired temperature will result.

The temperature of the air leaving the heaters is controlled by means of a second thermostat located in the room itself, and regulating the steam admitted to the heaters.

Hygrostatic Humidification Control

In certain instances it is permissible, during the summer season, for the dry bulb temperature to exceed that maintained in the winter, so long as the relative humidity is controlled at an approximately constant value. In such instances there is no provision for dehumidification, and the humidifier is used to effect as great a degree of cooling as possible by evaporation only. The dew-point temperature of the air leaving the humidifier then becomes the same as the wet bulb temperature of the outdoor air, the dew-point thermostat being inoperative. The dry bulb temperature in the enclosure is regulated in accordance with the prevailing wet bulb temperature of the entering outdoor air, being so controlled that the resulting relative humidity is that desired, and this regulation is accomplished by means of a hygrostat located within the enclosure usually adjacent to the thermostat which is used for winter control.

The shift from the room thermostat to the room hygrostat can be made either manual or automatic, as required. In most cases it is automatic.

The hygrostat, which is sensitive to relative humidity, controls the dry bulb temperature of the enclosure by regulating the volume of air admitted. This avoids the use of heaters, and takes advantage of the available sun heat, or heat from sources within the room. If the relative humidity within the enclosure is low, the dry bulb temperature being too high, the hygrostat opens the volume dampers and admits a greater volume of cooler saturated air from the humidifier. If the relative humidity is high, the dry bulb temperature being too low, the hygrostat reduces the volume of cooler air and permits the sun's heat, or the heat from sources within the enclosure, to restore the desired condition.

This method of control is widely used in textile mills.

Dew-Point Dehumidification Control

If, in summer, a dry bulb temperature lower than that of the atmosphere must be maintained, a dehumidifier is provided. In this case the dehumidifier acts, during the winter, as a humidifier, under dew-point control, and, during the summer, functions as a dehumidifier under the same dew-point control, except that the dew-point thermostat, at the dehumidifier, instead of regulating the steam to the spray water heater, regulates the temperature of the spray water delivered to the dehumidifier from the refrigerating equipment, or other source; and the room thermostat, instead of regulating the steam to the heaters, regulates the volume dampers in the supply ducts, controlling the temperature of the room by means of the volume of cooled, dehumidified air permitted to enter.

The two control instruments, then, regulate both the actual water-vapor content of the air and its dry bulb temperature, thereby fixing its relative humidity.

There are many variations of this control, but to a general understanding of air conditioning practice, a knowledge of this control is sufficient.

Addition of Moisture to Air

Unless some provision is made to add moisture to the air, the relative humidity of winter air, heated to ordinary indoor temperatures, is extremely low.

For example, outdoor air at zero, even if fully saturated, contains but 0.5 grains of water-vapor per cubic foot. If heated, indoors, to 70 deg., the capacity of the air to contain moisture is increased to 8 grains per cubic foot. If no moisture is added, the relative humidity of

the heated air indoors is but $\frac{0.5 \times 100}{8} = 6\frac{1}{4}$ per cent.

Air of such extremely low relative humidity exerts a tremendous drying effect upon human beings and upon the materials exposed to it. In the case of the human being, the effects upon the delicate mucous membranes of the nose and throat, as well as upon the skin itself, are extremely injurious, being the chief causes of our common winter ailments—coughs, colds, grippe, influenza, pneumonia, depression, lassitude.

Upon hygroscopic materials, such dry air exerts its powerful drying effect, causing all the phenomena incident to the loss of their moisture, such, for example, as the shrinkage and warping of furniture during the winter when the indoor air is artificially heated by the ordinary means.

The workability of many hygroscopic materials, such as textile fibres, as an especially important example, depends upon their moisture content, or regain, as it is called, which is proportional to the relative humidity of the surrounding air. Textile manufacture, continuing the example, which is one of many, cannot be carried on if the regain of the fibres is too low or too high. Therefore, provision must be made for conditioning the fibres, so as to establish and control the most desirable regain.*

The addition of moisture to the air is termed humidification, and the conditioning apparatus, when functioning to add moisture to the air, is termed a humidifier.

Pursuing the example of zero outdoor air, heated to 70 deg. indoors, and containing but 0.5 grain of vapor per cubic foot, corresponding to a relative humidity of 6¼ per cent, assume that it is desired to establish a relative humidity of 60 per cent in, say, the spinning room of a textile mill, the dry bulb temperature to be 70 deg. Since 70-deg. air requires 8 grains of water-vapor per cubic foot to saturate it, and it is desired to maintain a relative humidity of 60 per cent, the air in the room must contain 60 per cent of 8 grains, or 4.8 grains per cubic foot. By referring to the psychrometric chart, it is found that air contains 4.8 grains of water-vapor per cubic foot when it is saturated at 54½ deg. The dew-point, or saturation temperature in the humidifier must, therefore, be 54½ deg. In order to saturate the air at 54½ deg. so that it will take up the additional 4.3 grains of vapor required, it is necessary to heat the spray water until the temperature of the air leaving the machine is 54½ deg. (The air leaving a properly designed machine is always fully saturated). Thus, a humidifier is, in reality, a low-pressure, low-temperature boiler in which the water is evaporated into vapor and then caused to mix with the air.

In a humidifier the water acts as the medium which conveys heat to the air, and as the source of the water-vapor required to saturate the heated air.

When the temperature of the spray water is above that at which the moisture in the air will condense, the conditioning machine is functioning as a humidifier.

Removal of Moisture from Air

In many instances, on the other hand, the outdoor air of summer contains more moisture than is desirable, either impairing the quality of the product, or complicating its manufacture. For example, motion picture film is delicately sensitive to atmospheric moisture. If the film, in process of manufacture, is subjected to too moist air, it will become useless. In this, as in countless other instances, part of the moisture of the outdoor summer air must be removed.

Excess summer humidity is also the chief cause of human discomfort, and dehumidifying equipment must invariably be employed in air conditioning for human comfort and health.

When the conditioning machine is functioning to re-

* The term "regain" is somewhat misleading, without explanation. It does not necessarily mean that a material has gained moisture, because a given regain may be established by the loss of moisture. "Regain" means "moisture-content as referred to the dry weight of the material itself." Thus, a "regain of 10 per cent" means a "moisture-content of 10 per cent, referred to the dry weight of the material. Hence, ten pounds of dry cotton, when brought to a "regain of 10 per cent," would contain one pound of moisture, the total weight of the moistened cotton being eleven pounds.

CHICAGO INDOOR STADIUM—AN EXAMPLE OF AIR CONDITIONED BUILDINGS OF THIS SORT

move moisture from the air, it is called a dehumidifier, and the process of removing moisture from the air is termed dehumidification.

The removal of moisture from the air is accomplished by condensation, the temperature of the air being lowered below its dew-point, thereby causing the excess water to condense and fall into the tank of the conditioning machine. In this case the water acts solely as a conveyor of heat from the air (besides its cleansing action) and, as such, the finely divided mist is extraordinarily effective (practically 100 per cent).

Assume that it is desired to maintain the chocolate dipping room of a candy plant at a temperature of 65 deg. and a relative humidity of 50 per cent, when the outside dry bulb is 90 deg. and the outside dew-point, or saturation temperature, is 70 deg. If the outside dew-point is 70 deg., the air contains 8 grains of water-vapor per cubic foot. If it is desired to maintain an indoor relative humidity of 50 per cent at 65 deg. dry bulb temperature, the indoor air must contain 3.4 grains of vapor per cubic foot. (From chart.) The dehumidifier must therefore, condense and remove 4.6 of the 8 grains of vapor contained in the outdoor air. When air is saturated at 45 deg. it contains the required 3.4 grains of water-vapor per cubic foot. The dew-point, or saturation temperature maintained in the dehumidifier must, therefore, be 45 deg.

Since the entering outdoor air (or partly recirculated air) is above the dew-point temperature required in the dehumidifier, the spray water must be cooled. The cooling is accomplished by the admixture of naturally cold water from wells or lakes, or of mechanically refrigerated water.

The dew-point thermostat, when the conditioning machine is functioning as a dehumidifier, controls the temperature of the spray water (1) by operating a three-way valve in the suction line of the spray pump, thereby regulating the admixture of water from the settling tank of the dehumidifier, and from the cold water source, or (2) by regulating directly the refrigerating effect of a suitable refrigeration machine.

The Dehumidifier's Dual Function

The dehumidifier functions either as a humidifier or as a dehumidifier, without alteration or rearrangement, except that the valves in the control line from the dew-point thermostat are adjusted to connect the steam control to the water-heater for winter operation, and to connect the three-way mixing valve or the refrigeration control for summer operation.

Whether the requirement is humidification or dehumidification, the apparatus always operates under accurate automatic control, maintaining the required indoor conditions winter and summer, regardless of the outdoor weather.

3—PURITY

Pure, clean air is, of course, as important, both industrially and personally, as air whose temperature and humidity are properly controlled. Beside the filth and grime with which the air of cities and manufacturing centers is laden, countless mold- and disease-producing organisms are borne in the air. The dirt and other foreign matter, as well as mold-producing organisms, are disastrous to many processes of manufacture, while these and the organisms of disease seriously impair the efficiency and the health of the human being.

Properly designed conditioning machines, humidifiers and dehumidifiers, thoroughly wash and cleanse the air, removing practically all of the solid or soluble gas impurities, and most of the organisms of disease and decay.

The cleansing effect of the conditioning machine is produced by the finely divided and uniformly dense water spray and the staggered washer-eliminator plates against which the air is baffled as it leaves the chamber. These plates are flooded with water, so that their wet surfaces accomplish an extremely effective cleansing action.

In many instances it is essential that air distribution be soundless. It is not exaggerative to say that methods of air distribution adequate to any conceivable requirement of air conditioning, whether personal or industrial, have been developed and brought to high states of perfection.

4—AIR MOVEMENT

The effectiveness of any air conditioning apparatus depends as much upon the proper distribution of the air as upon the efficiency of the conditioning machine itself.

It may be said that an air conditioning installation is no better than its duct or distributing system.

To be effective, the conditioned air must be distributed uniformly over the entire area of the enclosure, and, especially in closed dry rooms or processing rooms, the circulation must not only be uniform, but vigorous.

The rate of effect of the air depends upon the rapidity of its movement against, and the intimacy of its contact with, the materials it surrounds. In drying, for instance, a certain minimum velocity of impact against the material must be maintained. This is necessary in order to insure penetration of the interstices in the structures of, or between, the material, and to dissipate the sluggish film of more nearly saturated and cooler air which surrounds the material as soon as drying begins.

Distributing systems must be carefully and scientifically designed to insure maximum effectiveness of the conditioned air which they deliver.

In many instances, especially in the conditioning of offices, studios, living quarters and such installations, it is essential that the air distribution be soundless. Means of accomplishing this have been devised after long, intensive study.

It is not exaggerative to say that methods of air distribution adequate to any conceivable requirement of air conditioning, whether personal or industrial, have been developed and brought to high states of perfection.

There are systems employing the principle of the ejector which meet, with high efficiency, many requirements beyond the suitability of direct low-velocity distribution.

For the conditioning of theatres, auditoria, stadia and

similar structures, which, contrary to general impression, require cooling during the most severe winter weather when occupied to 75 per cent or more of their capacity, systems of distribution by downward diffusion, from overhead supply openings, have been developed.

Evaporative Cooling

Due to the higher dry bulb temperatures of summer, the wet bulb depression is greatest during the summer season.

As has been explained, the wet bulb temperature is that temperature to which air would be cooled, by evaporation, if the air were brought into contact with water and allowed to absorb sufficient water-vapor to become saturated.

Thus, if outdoor summer air is drawn through a humidifier, wherein it will be completely saturated, its dry bulb temperature will be reduced to its wet bulb temperature, and the air will leave the humidifier at the outdoor wet bulb temperature. This cooling is accomplished entirely by evaporation, and is due to the latent heat required to convert the liquid water into water-vapor, such conversion occurring the instant the air is brought into contact with the mist in the spray chamber of the humidifier, the heat being taken from the air itself.

The spray water in the humidifier is used over and over, only that quantity being added which is actually absorbed by the air.

Thus, without any additional operating expense, a humidifier will, in summer, perform the function of cooling the air through the wet bulb depression.

The wet bulb depression is often, in some localities, as much as 25 deg. or even 30 deg., and quite commonly from 10 deg. to 15 deg., even in localities adjacent to great bodies of water, where the humidity is high and the wet bulb depression, therefore, correspondingly low.

In the vicinity of New York, for instance, the maximum outdoor wet bulb temperature is about 78 deg. On such a day the dry bulb temperature would probably be about 90, the wet bulb depression being 90—78=12 deg. In Denver, where the maximum wet bulb is usually less than 78 deg., the coincident dry bulb is usually much higher than 90 deg., resulting in a greater wet bulb depression, which means that more cooling can be accomplished by evaporation.

Consider a textile mill located in New England. Assume the spinning room to be 300 feet long by 100 feet wide, and that 500 motor horsepower is used to drive the spinning frames. All of the energy represented by the 500 horsepower is dissipated into the air within the mill as heat, being equivalent to 1,270,000 B.t.u. per hour, enough heat to maintain a comfortable temperature in the spinning room during the coldest weather.

In summer, if adequate ventilation, or some means of heat removal, were not provided, the temperature in the spinning room might exceed 150 deg.

In textile mills, as in many other industries, adequate ventilation with unconditioned outdoor air cannot always be arranged, because the movement of the tremendous air volumes requisite to such ventilation might cause drafts disastrous to fine production, not only because of the effect of the draft itself upon the fibres, but because of the dirt and dust resulting, and because the movement of such tremendous air volumes would make it difficult to maintain the proper condition of relative humidity within the room.

Evaporative cooling is the solution of such problems as this.

Assume an outside dry bulb of 95 deg. and a wet bulb of 75 deg., corresponding to a relative humidity of 40 per cent, the wet bulb depression being 20 deg. For efficient manufacture the spinning room should be maintained at a relative humidity of 60 per cent. If there were no air conditioning equipment, the room temperature would probably approximate 110 deg., with all the windows open and a fair breeze. The relative humidity of the outdoor air, entering at a temperature of 95 deg., and being heated to 110 deg., would be reduced to 26 per cent, a condition under which efficient manufacture is impossible.

In a mill properly equipped with air conditioning apparatus, the windows on the windward side of the building would be kept tightly closed, and the outdoor air would be drawn directly into the humidifier. In the humidifier the 95 deg. outdoor air would be saturated completely and cooled to the outside wet bulb temperature, or 75 deg.

The 75 deg. air from the humidifier would be uniformly distributed throughout the mill, without drafts, through the duct system. The quantity of such air would be regulated automatically so that, in absorbing the heat generated within the mill, its temperature would be increased 15 deg.

The room temperature would then be 75 deg. plus 15 deg., or 90 deg.;—20 deg. cooler than if no air conditioning equipment were provided, and 5 deg. cooler than the outside temperature!

Besides this reduction in temperature, the volume of air delivered to the room is automatically regulated so that the required relative humidity of 60 per cent is constantly and uniformly maintained.

Thus, it is possible, by scientifically applying the principles of evaporative cooling, to maintain a comfortable working temperature and a desirable relative humidity throughout the summer season, without the use of cold water or refrigeration.

Resumé

This brief, but fairly comprehensive, discussion has afforded, it is hoped, a reasonably clear conception of air conditioning as a tremendously important, widely adaptable, highly developed science, already definitely entered upon a career which will assuredly expand incredibly as time goes on.

The writer especially desires to emphasize that in conditioning for human comfort and health the function of the equipment during the long winter, though far less noticeable and spectacular than cooling and dehumidification during the short summer, is of even greater relative importance, a fact often overlooked or depreciated.

When it is realized that air conditioning, an unusually exact science, has either directly wrought or inspired improvement in every unit of the equipment utilized, and that many of these improvements have created entirely new and quite revolutionizing practices in wholly separate industries, then may be formed a fair estimate of its importance.

Refrigeration Piping

By R. C. Doremus

PIPE as pipe, is simply pipe, but when connected up in a pipe line, it becomes piping. As merchant pipe, it is just so much material, but when arranged advantageously, it can be put to work for a useful purpose. This is equally true whether it be in a heating system, refrigeration system or any system. It is commonly used for two general purposes, viz: (1) As so much surface for the transfer of heat and (2) as a conduit or conveyor for the handling of liquids, vapors and gases.

Analogy to Heating Systems

It will be the purpose of this article to consider it as surface for the transfer of heat—in refrigeration applications—and the question of mains will naturally follow as a later consideration. There is nothing mysterious about the piping of a refrigerating system for it is analogous to the piping of a heating system, except that the processes are in the reverse direction. In a steam heating system, the steam is condensed at the condensing surface or radiator, and in so doing gives up its latent heat of evaporation and changes its vapor state to liquid condensate. Similarly, in the direct expansion refrigerating system, the liquid refrigerant absorbs its latent heat of evaporation at the coils or evaporator, and in so doing changes its liquid state to vapor.

In the hot water heating system, inasmuch as the temperature of the water is higher than the surrounding air, heat will flow downhill from the water through the pipe or radiation surface to the air and the water will cool down while the air warms up, due to the exchange of heat units and the natural tendency to equalize in temperature. This differs from the steam heating system in that the water simply gives up some of the heat of the liquid. Similarly, in the brine refrigerating systems, inasmuch as the temperature of the brine is lower than the surrounding air or material, heat will flow downhill from the air to the brine through the coil or surface and the air will cool down while the brine warms up, due to the temperature gradient. This differs from the direct expansion system in that there is no latent heat of evaporation, but simply an increase in the heat of the liquid, or sensible heat as it is sometimes called.

In the hot water heating system, the water is used as a conveyor of heat from the hot water heater or boiler to the radiator, and in its continuous circulation it is alternately heated at the heater and cooled at the radiator over and over again. In the brine refrigerating system, the brine is also a heat conveyor and in its circulation it is heated up by the heat transferred from the air or material through pipe coils to the brine and must, therefore, be cooled by the direct expansion of a refrigerant in a brine cooler or evaporator before it is circulated through the coils again to pick up another load of heat. Water might be used as the conveyor, except that it would freeze to ice when cooled by the direct expansion cooler and hence the freezing point is lowered by the addition of sodium or calcium chloride to form brine.

Choice of Refrigerant

The choice of whether a certain refrigerating system is to be a direct expansion system or an indirect brine system is usually a question left to the consulting or designing engineer. Very often a problem is such that if a direct expansion system were used a slight leak of refrigerant would be readily absorbed in materials stored or would be obnoxious in air and cause people to become panicky; therefore, brine systems are frequently used in theatres and auditoriums and cold storage of eggs, etc. The storage of potatoes, onions, nuts, and such commodities can be handled very well with direct expansion, for they are very slightly affected, if at all, by a minor ammonia leak. There are many phases to be considered in such a choice if the system is to be satisfactory to men and material.

The refrigerant chosen must have favorable physical and chemical properties, and, above all, must condense with natural air or water temperatures at pressures that are not excessive and must have evaporating temperatures at convenient suction pressures, preferably slightly above atmospheric pressure to avoid compressor stuffing —box leakage.

Fig. 1 shows a comparison of some of the more frequently used refrigerants including ammonia, carbon dioxide, sulphur dioxide, ethyl chloride and methyl chloride.

Fig. 1
Comparison of Refrigerants

	NH_3	CO_2	SO_2	C_2H_5Ce	CH_3Ce
1. Heat content of vapor in B.t.u. @ 5° F.	539.9	101.0	162.3	165.4	163.7
2. Heat content of liquid in B.t.u. @ 86° F.	60.6	46.8	18.2	23.1	25.1
3. Refrigerating effect per lb. in B.t.u.	479.3	54.2	144.1	142.3	138.6
4. Lbs. per minute per ton of refrigeration	0.417	3.69	1.39	1.41	1.45
5. Specific volume of saturated vapor @ 5° F. in cu. ft.	8.20	0.267	6.66	17.06	4.72
6. Displacement per min. per ton in cu. ft. for operation 5°-86° F.	3.42	0.986	9.25	23.95	6.83
7. Mean effective pressure 5°-86° F. in lb. per sq. in.	66.6	435.0	24.4	8.75	3.65
8. Pressure in lb. per sq. in. absolute @ 5° F.	33.79	334.2	11.8	4.65	21.0
9. Pressure in lb. per sq. in. absolute @ 86° F.	170.2	1039.0	66.45	27.1	97.5
10. Horsepower per ton of refrigeration 5°-86° F.	0.995	1.87	0.985	0.915	1.07

Our future discussions of refrigeration piping will be largely devoted to the systems using ammonia direct expansion and calcium chloride brine.

FIG. 2

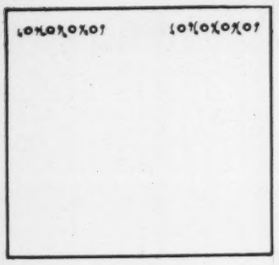


FIG. 3

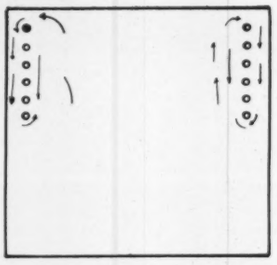


FIG. 4

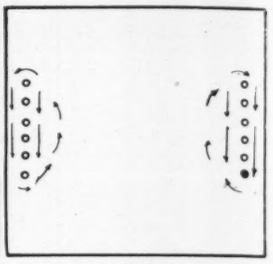


FIG. 5

Location of Refrigerating Surfaces

The arrangement of piping for refrigerating surface often spells a relative degree of success for the whole system. In heating systems, the coils are located at the bottom of a room for gravity air circulation, inasmuch as this is the point where the coolest air comes in contact with the heating surface. Likewise, in refrigerating a room by gravity air circulation, the cold coils must be located near the top of the room for here the warmest air comes in contact with the cooling surface, is cooled, drops to the floor, is heated by refrigerating the material in the room and rises to the ceiling coils with its load of heat, where it is again cooled. A brief study of coil arrangement will be beneficial in order to understand desirable designs and installations of coils in refrigerator boxes, coolers and freezers.

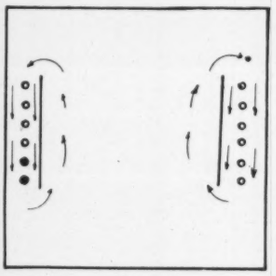


FIG. 6

The pipe surface shown in Fig. 2 is arranged in one single flat coil covering the whole ceiling. This is better adapted where the ceiling is the roof of a building having direct sun radiation, but does not produce a vigorous circulation.

Fig. 3 shows two single flat coils at ceiling and likewise produces circulation only at ceiling, whereas goods are stored on floor.

Fig. 4 shows single flat coils at side walls near ceiling and circulation is local.

Fig. 5 shows single flat coils at side walls a little lower down and nearer the goods, but circulation is local and ceiling may be wet from condensation.

Fig. 6 shows single flat coils at side walls like those in Fig. 5, but with an apron which aids air circulation. It also will have a wet ceiling.

Fig. 7 shows single flat coils high upon side walls with bunker pans and aprons. This improves circulation and has advantages over other bunker types where headroom is limited.

Fig. 8 shows a wider room with double bunker. Single bunkers should not be used in rooms over 9 ft. wide if the height is approximately 9 ft. With this arrangement the coldest air is at side walls and coil aprons are needed to induce circulation on account of heat infiltration at walls.

Fig. 9 is a better arrangement than Fig. 8 because heat infiltration through walls warms air in room next to walls and thus assists in providing more vigorous air circulation.

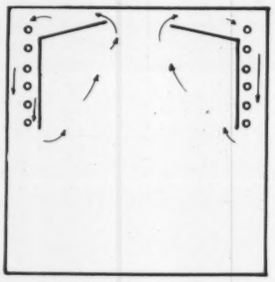


FIG. 7

Fig. 10 shows a single bunker with a direct expansion coil and a congealing or holdover brine tank also containing ammonia pipe coils. The congealing tank assists in holding the temperature of the air in cases of intermittent compressor operation.

Fig. 11 shows an arrangement similar to that in Fig. 10, except that brine tubes are used as holdover capacity. These tubes are galvanized iron tubes containing an inner ammonia pipe and the space between the pipe and tube is filled with a brine of correct strength, so that it will congeal during periods of compressor operation, but will melt at a few degrees below air temperature. In this manner it takes up its latent heat of fusion of approximately 100 B.t.u. per lb.

Fig. 12 shows a bunker arrangement for a very wide room of comparatively low temperature.

Fig. 13 shows another arrangement for a very wide room of higher temperature than Fig. 12.

Fig. 14 shows two single flat wall coils and a single central bunker double coil with pan. This is a good arrangement for storage of celery, etc., where goods are

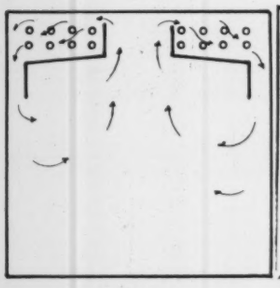


FIG. 8

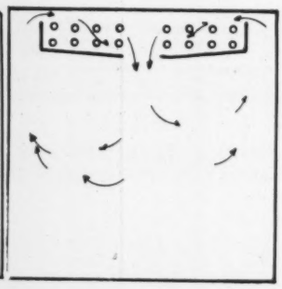


FIG 9

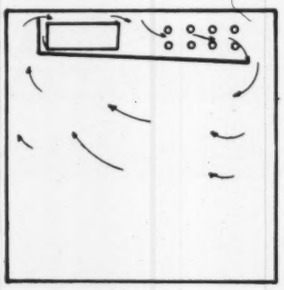


FIG. 10

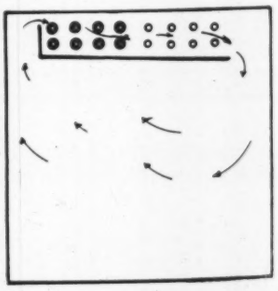


FIG. 11

bulky and central bunker coil can be arranged over aisle.

Fig. 15 shows two single flat wall coils with aprons and a single flat ceiling coil without pan. This is not a good arrangement for a cooler on account of local circulation and spoilage of goods through coils dripping. It is more typical of a high temperature freezer. A colder freezer may be had by doubling the ceiling coil.

Fig. 16 shows a typical double bunkered cooler from 9 to 15 feet wide where headroom is not limited.

Fig. 17 shows a typical narrow freezer room with a heavy bunker coil such as will be found in a sharp freezer or an ice cream hardening room.

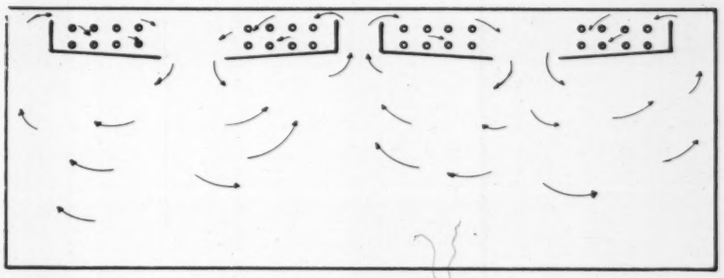


FIG. 12

Air Circulation Essential

Inasmuch as the air convection currents in the refrigerated space are the conveyors of the heat load from the goods to the coils, it is essential that the circulation be thorough and rapid across the coil surface. If it is fairly rapid the heat transfer through the coils is at a good rate, whereas if it is poor the rate of transfer is poor, and more coils must be used. Circulation is essential to reach the heat in all of the goods stored. A third benefit of air circulation is to equalize the air temperature in the room which, in good design, will not vary more than 3 deg. or 4 deg., but in poor design may amount to 15 deg. to 18 deg. Inasmuch as the primary function of a cold storage is not one of cooling but rather of holding the goods at a constant temperature over long periods of time, a big temperature gradient in a room is an admission of failure. Another result of good air circulation is that moisture in the room is deposited as frost on the coils, and this frost frequently holds odors with it. Thus, it purifies the air by entraining the odors. The moisture would otherwise be deposited on the walls and goods if the refrigeration was by radiation alone rather than by a combination of radiation and convection.

With some commodities, such as eggs, the piping coils dry up the air too much, due to this faculty of depositing the moisture, and it becomes necessary to use a water sprinkling can or humidifier to "water the eggs."

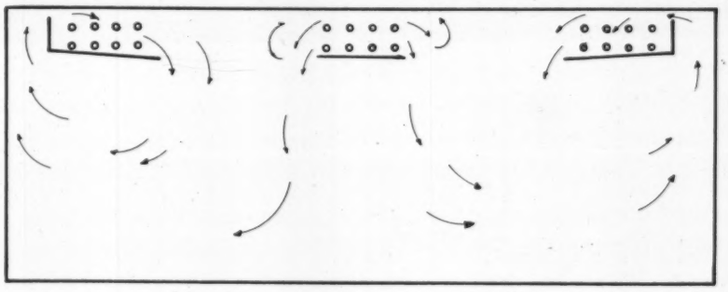


FIG. 13

The best method of piping a cooler or freezer in general cannot be definitely stipulated in any rule, for each room is a problem by itself and is governed too much by:

1. Kind of goods stored.
2. When stored and how fast.
3. The method of storing goods.
4. Size of room.
5. Insulation value.
6. Adjacent rooms or sun's radiation.
7. Frequency of door operation.
8. Mean temperature desired.
9. Humidity desired.
10. Whether short or long period.

In general, it is preferable to plan the coils in bunkers or lofts with aprons, pans and baffles well insulated to induce a positive, vigorous circulation. In the case of wall coils, an insulated apron is frequently needed also to prevent freezing of adjacent goods through direct radiation. In the case of freezers and coolers located next to each other on the same floor, it sometimes happens that goods freeze in the cooler adjacent to the common wall even though an insulation belt is installed between the two rooms. This can be prevented by the installation of a single flat wall coil on the cooler side of the insulated wall through which warm brine is circulated. Thus, a warm brine screen is formed to prevent this radiation.

Avoid Obstructions

The effect of obstructions in the air circulation should be avoided—the hot air and cold air ducts should be ample, coils should not be too close to one another as to frost shut and stop air passage, and changes in direction of air flow should be gradual. Stream flow practice should be followed to induce circulation rather than hinder it. The coils should be defrosted frequently to avoid heavy accumulations of frost for they insulate the surface against quick transfer of heat. True, these are construction and operating points, but they are vitally essential, for the pipe alone, without these contributing factors, cannot do its work efficiently.

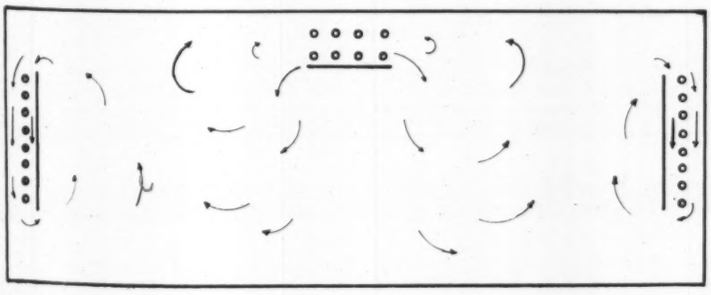


FIG. 14

It should be remembered that air is caused to circulate in a gravity room by the difference in densities at different temperatures—the warmer air rises to the ceiling where it passes over the refrigerating pipe coils, becomes cooled, drops to the floor due to gravity and displaces warmer, lighter air, which rises to the ceiling to be again cooled by the coils. In general, the more vigorous the circulation the better the room will operate, which emphasizes the care needed in doing everything to induce good air circulation.

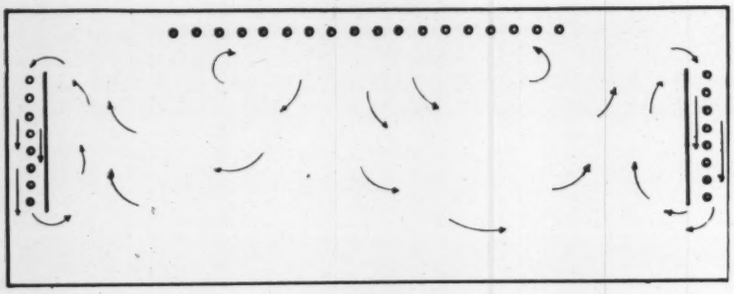


FIG. 15

There are, however, some commodities, such as cut meat, which will dry up and "burn" through too vigorous a circulation of dry air. In such cases, the circulation should be only moderate, or, if vigorous, humidifiers should be installed. In some instances of the sort, the piping has been eliminated entirely and the refrigeration accomplished by brine spray bunkers. This system has met with good success in large meat packing plants for it keeps the relative humidity high and results in an elimination of "case hardening" the meat, bone rot and shrinkage in weight, thereby improving the market value.

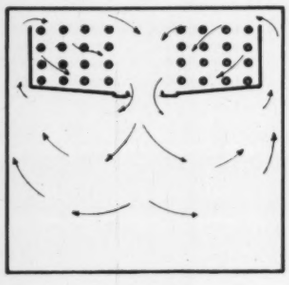


FIG. 16

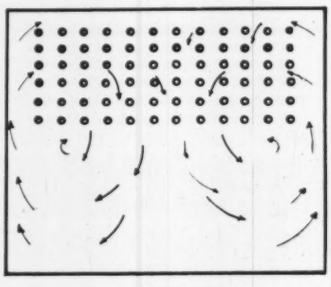


FIG. 17

Gravity air circulation is only applicable to freezers and small or medium size coolers. Medium size coolers are frequently piped with coils augmented by brine spray bunkers or brine sprays in false wall ducts, thus improving the air circulation. In large coolers a vigorous air circulation is obtained by a forced air ventilation system.

Forced Circulation

In this case, the location of the pipe coils is not so important, for gravity plays a very small part when the currents of air are forced by mechanical means. The bunker piping will be placed in an adjacent room from floor to ceiling, so arranged that the velocity of air over the coils is high and the air will not "short circuit" but will scrub the coil surface efficiently. Then the cold air is led into the room in a supply duct of galvanized iron with frequent adjustable vent louvres and the exhausted or displaced air taken from the room in a similar duct back to a fan which recirculates it through the bunker room, causing a constant circulation, as shown in Fig. 18.

This system requires less pipe surface than the gravity system due to higher air velocity over the coils and consequently higher heat transfer. This means that it gathers frost and odors faster and must be defrosted periodically. Inasmuch as a duct system has been installed, it is a simple matter to arrange a defrosting bypass to blow the bunker air to the outside atmosphere instead of back into the room where the goods are stored, and this further assists in vitalizing the air in the room and preventing odors.

In some coolers, an arrangement of pipe coils and ducts are used as indicated in Fig. 19. This shows the inlet duct with distributing louvres and with pipe coils on both sides, all located over the aisle. The outlet ducts are at each side. This has the advantage of positive air circulation, high velocity over coils and requires only space for a fan instead of a bunker room. In defrosting, the odors cannot be blown out of the room as in the former case, but they are imprisoned in the snow and frost and removed from the room in this manner.

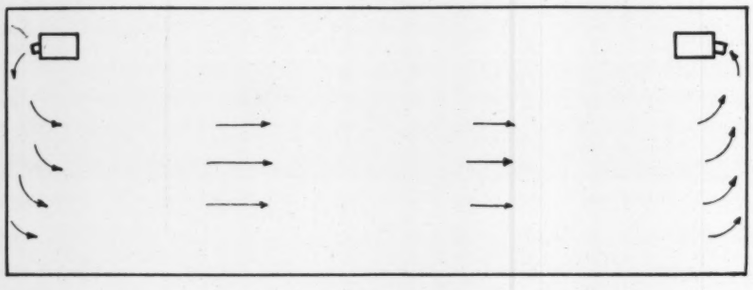


FIG. 18

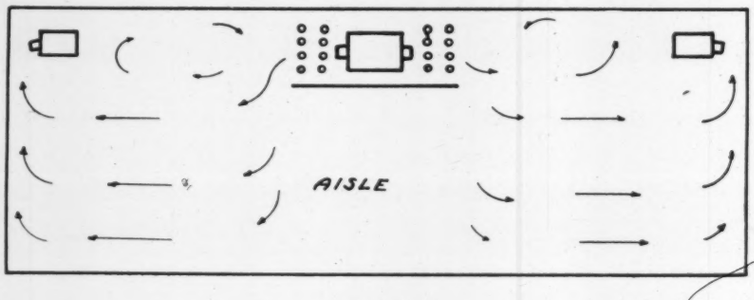


FIG. 19

Questions pertaining to the quantity of piping required for direct expansion and indirect brine systems, as well as their mains and methods of distribution will be considered in a later issue of this magazine.

Making the Unit Heater Fit the Job

By C. D. Graham

TEN years ago, or for that matter five years ago, it was not unusual for the unit heater representative to be asked, "What is a unit heater?" Some engineers and plant executives did not know whether a unit heater was a new type of boiler or a new device which required no steam generating plant.

During the last few years unit heaters have made such rapid progress that it is now difficult to find a man even remotely interested in factory layout, maintenance or production who is not cognizant of unit heaters and the savings they can effect. In short, unit heaters have been so generally accepted as an economical means of heating industrial buildings that the purchaser is often prone to accept units of any make or type and consider his problem solved. In so doing, he fails to consider that each building in a plant may offer an individual heating problem. To secure the best results throughout a group of buildings, a careful engineer must, therefore, go further than the mere decision to heat by the unit method. He must equip each building with the type of unit best suited to its particular need.

It is not infrequent to hear a certain make and type of unit highly praised by one man and the same make and type unfavorably criticized by another. Were the units from the plant of the satisfied and dissatisfied users interchanged, the opinions of the two men would probably not be affected, as the real basis of complaint is usually improper application rather than the mechanical operation of the units.

Major Types of Unit Heaters

There are four major types of unit heaters—first: high velocity, floor mounted; second: high velocity, suspended; third: low velocity, floor mounted; fourth: low velocity, suspended. True, some engineers carry the classification still further and divide any or all of the four above major types into "draw-through" and "blow-through" designs. This sub-division is a mere detail of construction and need not, therefore, enter into this discussion.

General Rules for Fuel Economy

When choosing heating equipment, the purchaser or his engineer should weigh first cost against operating cost, bearing in mind that the latter is of vital importance. For the best fuel economy on the usual heating job, where a definite working zone temperature is desired, there are several general rules which apply.

I—Since cool air is heavier than warm air, it naturally settles at the floor line; therefore, the relatively cool air should be removed from the floor, allowing warmer air from above to replace it.

II—The natural tendency of warm air is to rise. A high discharge velocity acts in two ways to prevent this. The energy of the air stream holds the warm air closer to the working zone and causes an induced circulation of room air, which mixes with the warm air stream, thereby reduc-

FIG. 1.—A TYPICAL, HIGH VELOCITY, FLOOR MOUNTED UNIT HEATER INSTALLATION

ing its temperature and, consequently, reducing its tendency to rise.

III—Obviously, a low final temperature is desirable, since the air stream will have less tendency to rise. Considered from the standpoint of fuel economy alone, it would be far better to heat with air at 80 deg. or 90 deg. than to heat with air at 150 deg. or 160 deg., but the interest on the initial investment renders heating with extremely low final temperatures inadvisable. It is impossible to state exactly what final temperature should be used, but, in general, a discharge temperature of more than 70 deg. above room temperature should be avoided.

Rules one and two, above, suggest, for many industrial buildings, high velocity floor mounted units, to obtain operating economy. There are, of course, cases where local conditions make it unwise to use this type of unit. For instance, in an existing plant where the return lines are overhead, the cost of revising the piping may make the use of floor mounted units impractical. If such is the case and suspended units must be used, the high discharge velocity is suggested by rule two. The advantages of rule one may be included if recirculating ducts are used to remove the cooler air from the floor line.

Occasionally, conditions demand that fuel economy be sacrificed to some extent for quietness of operation. Examples of this are garage show rooms, auditoriums, etc. A number of industrial buildings require only relatively quiet operation. Here, the extremely low velocities (800 feet per minute or below), essential for absolute quietness, should be raised to about 1000 or 1200

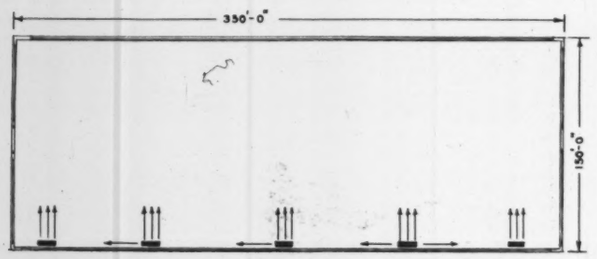


Fig. 2.—This Layout of High Velocity, Floor Mounted Units, with All the Units Along One Wall, Simplifies the Piping

feet per minute (1500 and above is generally considered high velocity).

In general, low velocity suspended units are built in relatively small sizes. Consequently, they are frequently used in shop offices, toilets, etc., where the heat losses are small. Here, fuel economy is not vital, since the capacity is negligible compared with the main section of most plants.

The foregoing discussion of the various types of unit heaters serves to bear out the statement that the engineer must go further than the mere decision to heat by the unit method. There is always one type of unit best suited for a specific heating job. Not infrequently, a well-balanced heating installation requires units of more than one type. This is especially true of plants occupying several buildings which are not identical in structure or use.

Selecting Unit Heaters for Specific Case

To make the subject more concrete, let us take an imaginary specific case. The Jones Manufacturing Company of Chicago is building a new plant and warehouse in the east. The main manufacturing building will be 350 feet long by 150 feet wide by 36 feet high, and is to be devoted almost entirely to automatic machinery. The finishing building will be 150 feet by 75 feet wide by 15 feet high, and will employ, almost entirely, women seated at light noiseless tasks. The warehouse will be

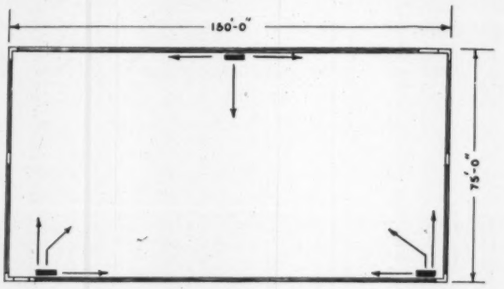


Fig. 3.—Floor Mounted Unit Heaters Arranged in This Manner Give Excellent Distribution

100 feet long by 75 feet wide by approximately 30 feet high at the peak and will require only sufficient heat to prevent the sprinkler system from freezing.

Obviously, it would be possible to purchase a sufficient number of almost any make or type of unit heaters to heat these three buildings. A moment's consideration, however, will readily demonstrate that these buildings should be considered separately.

The main manufacturing building covers a relatively large area and, therefore, offers an excellent opportunity for the application of the high velocity, floor mounted type of unit. Before making a definite decision to this effect, the problem should be considered from all angles.

First, consider noise. It is only natural that high discharge velocity units would be more noisy than low discharge velocity units. In this building, which will

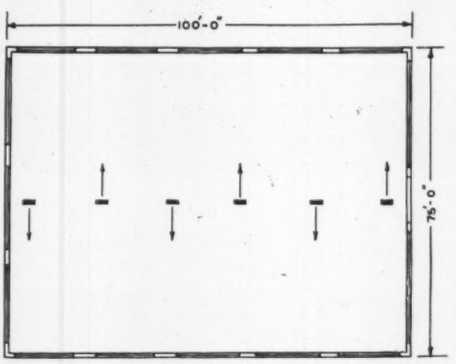


Fig. 4.—There Are a Number of Possible Satisfactory Locations for These Suspended Units, but the One Shown Combines Good Distribution with Simplified Piping

house automatic machinery, any noise from a unit heater would not be noticeable.

The next point to consider is final temperature. In this plant, steam will be generated at about 90 pounds pressure. Were standard units of most makes to be

used on this pressure the final temperature would be excessive. There are two ways of avoiding a high final temperature—one is the installation of pressure reducing valves, thereby rendering the entire heating system a low pressure installation. The other possibility is the use of units having coils smaller than standard. Either of these two methods of reducing the final temperature will give fuel economy. The final decision as to which should be used depends largely upon local conditions. Where exhaust steam is plentiful, it would be more economical to use a low pressure system and supply enough make-up steam through a pressure reducing valve to supply any deficiency in exhaust steam. Where exhaust steam is not available, it would be more economical to use units having reduced coils, thereby getting the savings in installation costs which accompany high pressure distribution on account of the smaller pipe sizes.

In this particular building there will be little exhaust steam available; therefore, units having small coils are selected. The larger the capacity of a unit, the less the cost per thousand B.t.u. for a given make and design. It is, therefore, economical to buy a small number of large units to deliver a given B.t.u.

Hence, the selection for the main building is high velocity, floor mounted units, each unit having relatively large capacity. The sketch in Fig. 2 shows a layout for this building.

In the finishing building, the conditions are somewhat similar, the principal difference being that noise would be objectionable, since the work performed in this building is relatively quiet. The units for this building would differ from those for the main building in that they will have lower outlet velocities (1000 to 1200 feet per minute). For uses such as this, it is sometimes found advisable to use units equipped with two-speed motors. All heating equipment is (or should be) selected to meet the maximum demand; therefore, the maximum speed (which may be slightly noisy) would be required only during heating-up periods or during extremely severe weather; at all other times the lower or quiet speed would be available. Fig. 3 suggests a good installation for this building.

Now, let us consider the warehouse, where heat is required to prevent the sprinkler system from freezing. At first thought, it would appear that floor mounted, high velocity units would be an economical selection. But, in this building, heat is not essential at the working level; therefore, it would be a waste to install equipment which would hold heat in the working zone. The selection, therefore, is low velocity, suspended units without deflectors, for this type of equipment will allow the maximum of heat to bank at the ceiling, where it is required. (A high discharge temperature is satisfactory here.) See Fig. 4.

By using this type of unit in the warehouse, fuel would be saved, since the lower portion of the building would not be heated to any great extent.

The application of thermostatic control to all the units will effect still further savings. Few owners realize that from 10 to 30 per cent of their fuel is wasted by overheating and through open windows, when they rely upon workmen to regulate the heat manually. Unit heaters are well adapted to thermostatic control by starting and stopping their motors. The cost of thermostatic motor controls is usually saved in less than a year.

Sprinkling Systems for Airplane Hangars

REPRESENTATIVES of leading airplane manufacturers and airplane operators have opposed recommendations of the National Board of Fire Underwriters for sprinkling systems in hangars exceeding 10,000 square feet of floor capacity.

The National Board of Fire Underwriters has not, it says, taken the stand that sprinklers must be installed in airplane hangars, but its attitude has been that, based upon experience with automatic sprinklers in similar occupancies, such means of protection would, in the opinion of that organization, be effective and efficient in hangars and should be accorded such recognition. Since to date no sprinklered hangars have been constructed and therefore no actual experience of sprinklers in hangars has developed, the aeronautics industry has objected to sprinkler recognition principally on the grounds that water is not a suitable extinguishing agent for the character of fires likely to be encountered, and that the arrangement of airplane storage would also prevent the extinguishment by this means. This difference of opinion has led to an offer on the part of the National Automatic Sprinkler Association to conduct tests under actual conditions to demonstrate the value of sprinklers. The National Board of Fire Underwriters has agreed to postpone promulgation of its regulations pending the making of these tests.

In this connection the Department of Commerce has organized a committee to arrange and conduct the tests. Following is a list of the members of the committee and the organization by which they were designated.

H. E. Blee, Chairman, Aeronautics Branch, Department of Commerce.
W. S. Garland, Bureau of Aeronautics, Navy Department.
R. W. Hendricks, Underwriters' Laboratories.
Ira G. Hoagland, National Automatic Sprinkler Association.
Major Frank M. Kennedy, United States Army, Air Corps.
W. Laurence LePage, Aeronautical Chamber of Commerce of America.
N. D. Mitchell, Bureau of Standards.
H. E. Newell, National Board of Fire Underwriters.
Starr Truscott, National Advisory Committee for Aeronautics.

The Bureau of Standards in addition to naming a representative on the committee has agreed to furnish such personnel and instruments as may be necessary for making a permanent record of tests. It is expected that the U. S. Army will furnish test facilities in the shape of a hangar and obsolete planes. No date has yet been set for the tests.

Planning Air Filter Installations

By W. G. Frank

WHEN the air filter was a newcomer in the air conditioning field—and not so very long ago this was the case—little thought was usually given to the proper way of installing it. After the decision had been reached to incorporate one of the new air cleaning devices in a ventilating system, a hasty survey would be made to find a space somewhere near the fan that would accommodate the filter. After much cutting and trying a place would finally be found and the filter drawn into the plans. In this process the air filter engineer usually had to relinquish step by step most of his ideas as to what would constitute a good filter layout, providing he wanted to see a filter used at all. Frequently then, when the system was finally placed in operation, the air filter performed only after a fashion, rather than in the way the manufacturer had intended.

Today, the basements of modern buildings have become so crowded, that no amount of ingenuity will find a place for the comparatively large air filter unless its layout is taken into account from the very start. Experience has shown that it pays to study the requirements of air filters since they have proved their economical value beyond any doubt, and are considered a vital part of ventilating and air conditioning systems.

With this in mind suggestions are here given which will aid in planning an air filter installation. They refer especially to the type of filter known as "viscous coated." The newly developed dry-type of filter operates on different principles. Its installation is subject to different rules and will require a separate discussion.

The Right Amount of Filtering Surface

A filter, to work satisfactorily, should have a sufficient filtering area for the amount of air it is intended to handle. Such a statement sounds so elementary as to be hardly worth mentioning. However, it is no uncommon experience to find an air filter laboring under an overload varying from 20 to 60 per cent—or even more. What is the explanation?

When a ventilating system is laid out, the capacity of the air filter is usually specified to correspond with that of the fan. But frequently, after the building is finished and occupied, the actual requirements, for various reasons, are found to differ from those originally intended. When the occupants of parts of the building change, rooms are often utilized in a way different from that intended in the original plan. Or, in the warm season, the ventilating system is called upon to produce a cooling effect, which means an increase in capacity. It is an easy matter to speed up the fan and obtain a larger volume of air. But the filter remains the same and is suddenly found to be inadequate, or nearly so, to take care of the increased air volume.

How will such an overload affect the performance of the filter? The resistance increases rapidly under an overload, the rate of increase depending upon the type of filter medium used. An added load is thereby thrown on the fan. As the velocity of the air within the filter medium increases, its cleaning efficiency is naturally affected. The carefully balanced action of dust particles in the filter screen, which lends them enough momentum to leave the air stream and come into contact with the surface of the medium, has been partly destroyed. The very small dust particles which would have been impinged against the viscous coated surfaces, are carried along as the air flow straightens out. Some filter media are more sensitive to this change than others but for each a limit can be found where its effectiveness begins to suffer.

The same characteristic change takes place when the air volume is greatly below the rated capacity. However, the detrimental effects of overload are much more pronounced than those of underload. It is generally accepted that an excess in capacity of 10 per cent over the rating of the air filter is the limit in this direction, whereas there is more flexibility in the way of underload.

It might be well here to call attention to the basic difference between dry and viscous type filters. The

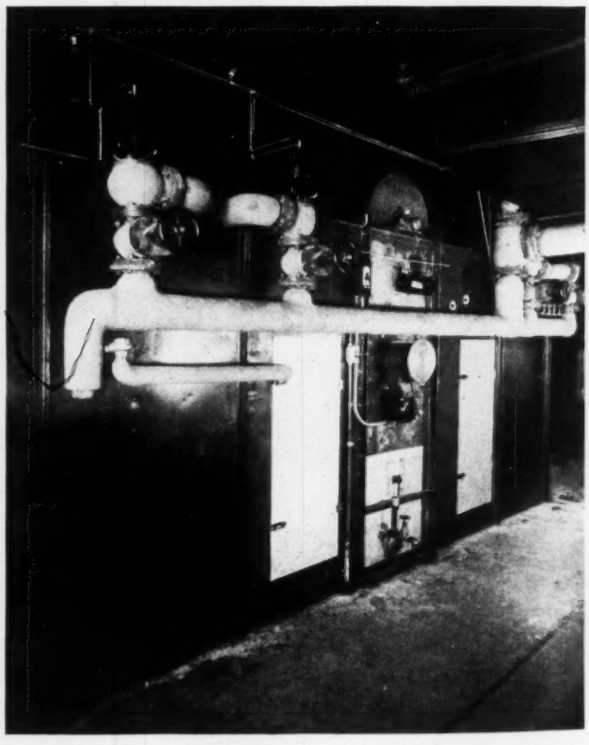


FIG. 1.—SELF CLEANING VISCOUS FILTER (MOTOR OPERATED) INSTALLED BETWEEN TEMPERING AND HEATING COILS. AREA AND OVERALL DIMENSIONS OF FILTER AND COILS ARE ALIKE AND CONNECTING SHEET METAL WORK EXTENDS IN A STRAIGHT LINE. DIRECTION OF AIR FLOW FROM LEFT TO RIGHT

openings in the filter screen of the latter type are very much larger than the dust particles it will retain from the air flowing through it. Filter media of the dry form are usually so dense as to restrict the passage of dust particles by virtue of openings smaller than the particles. While they both react alike against overload, the dry filters are very much favored by any reduction in air velocity.

It should be further remembered that an overload amounting to one-half of the original rating will also carry 50 per cent more dust into the filter. If the dust content of the air is quite heavy, the result will be a severe strain on the performance of the filter. If cell filters are being used, it may be necessary in such a case to remove and clean them so often that the maintenance of the air filter is far from economical.

Automatic filters are usually flexible enough to take care of this phase of an overload condition in that they can be adjusted to operate for longer periods of time, or more frequently. However, unless their auxiliary devices, such as soil clarifying and sludge handling equipment, are enlarged or speeded up, these are likely to suffer.

The Layout of Air Filter Installations

Next in importance are the arrangement of the filter in relation to the air flow, and the shape of the connections to and from the filter. The observance of a few simple rules may mean the difference between complete or partial success of an installation. Basically there is no difference in this respect between the requirements of filters and other airhandling equipment. But filters are just a little more sensitive and suffer particularly under wrong air distribution, eddy currents and dead air corners.

The customary air velocities in ducts are several times larger than those prevailing at the face of the air filter. The connections, therefore, between the filter and the duct system or fan inlet should be sloped as gradually as possible. Best results will be obtained if the incoming air enters the filter chamber at right angles to the filtering area. An attempt should be made to have the face of the filter of nearly equal height and width. In very tall and narrow filters or low and wide shapes, some part of the filtering area might be ineffective, which, in turn, produces an overload on the remaining area. Filters arranged diagonally, or parallel to the air flow, usually suffer for the same reasons.

If space conditions are cramped and a uniform air distribution cannot be obtained by ordinary means, provision should be made for baffles or vanes to direct the air flow in the desired way. Their position can be determined only approximately in advance and not too much faith should be put in "trained arrows," although they may look very good in a drawing. It is best to provide some adjustment for the baffles and find their best position by measurement with an anemometer after the system is operating.

It should be made a rule to provide sufficient space in front as well as behind the filter to make it accessible for inspection and service. A distance of two feet may be regarded as the minimum. Three feet would be better. Access doors of convenient size should be provided in

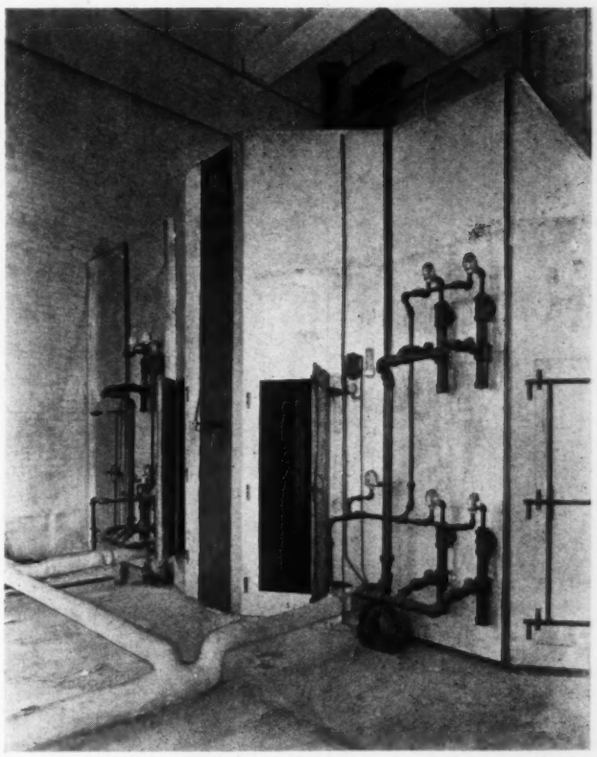


Fig. 2.—Self Cleaning Viscous Filter (Hand-Operated) Installed Between Tempering and Heating Coils. Sheet Metal Connection Slopes Gradually from Filter Towards Both Coils. Note Access Doors. Air Flow from Left to Right

the sheet metal connections leading to and from the filter. The door on the clean air side should be lined with felt to prevent infiltration of outside air. For the same reason, all connections of the clean air ducts should be as air tight as possible. This point deserves a great deal of attention. Consider that it is just as important and at the same time more difficult for the filter to extract the last few per cent of the dust as it is to eliminate the great bulk of it. It would be poor policy, therefore, to let small amounts of outside air get into the clean air and so greatly handicap and, even destroy, the effectiveness of the filter.

By all means an electric light should be installed in the chamber in front of and behind the air filter. These chambers are usually cramped and drafty, and the side of the incoming air will besides collect a great deal of atmospheric dust after a short time of operation. Electric lights, which can be turned on from the outside, will encourage the operating personnel to make more frequent and thorough inspections of the filter.

Distance from Heating Coils

The logical place for an air filter in a ventilating installation is as near the air inlet of the system as possible. In this location, the filter will protect the entire system from the atmospheric dust. As a rule, we find the heating coils are also placed near the air inlet and can combine the installation of filters and coils. This makes for an easy and simple connection as the recommended air velocities over the face of coils and filters are nearly the same. It economizes on space as well as

static pressure drop. The velocity of the air coming from the inlet duct is reduced to about 500 ft. per minute. After passing through heating coils and filter, the cleaned and heated air goes into the fan at gradually increasing velocity.

In analyzing the place which the filter should have in relation to the heating coils, we will have to look into the natural requirements of a viscous type filter. The more uniform the conditions under which the filter operates, the better will be its performance. Severe changes in temperature will affect the characteristics of the filter oil which is, we might say, the life-blood of the viscous type filter. Filter oils are mineral oils and are found to thin out in very high temperature and to thicken under low temperatures. While this is an inevitable law, all ordinary changes in temperature will not impair the usefulness of filter oils which are specially selected and treated for use in air filters.

In central fan systems, where all heating and ventilating is accomplished through the fan system, the heating surface is usually subdivided in "tempering coils" and "heating coils." In all cases where a filter is used, it is desirable that these two sections be set a distance apart and the filter placed in between. The tempering coils will then protect the filter from the extreme cold of the winter.

Where the heating of the building is done by direct radiation, the air handled by the fan is heated to approximately room temperature before it is distributed throughout the building for ventilation purposes. In this case the filter should be located between the heating coils and the fan. Where the air serves partly to heat the building, and in all cases where preheating and reheating coils have to be placed close together, it is an open question as to where the filter should be located. If placed between the coils and the fan, the heat from the coils increases the temperature in the filter chamber to considerably over 100 deg. fahr., especially when the fan is shut down. In this case it is usually advisable to place the filter ahead of the coils and provide tempered air for the filter by opening the door of the filter chamber on the incoming air side.

Outdoor Installations

Filters for ventilating installations are not usually located outdoors without the protection of pent houses and heating coils. However, in many industrial installations and in practically all applications for compressor and Diesel engine service, filters are placed wherever space is available, usually outdoors. Such filters should be protected from the weather by small housings which are frequently built integral with the filter structure

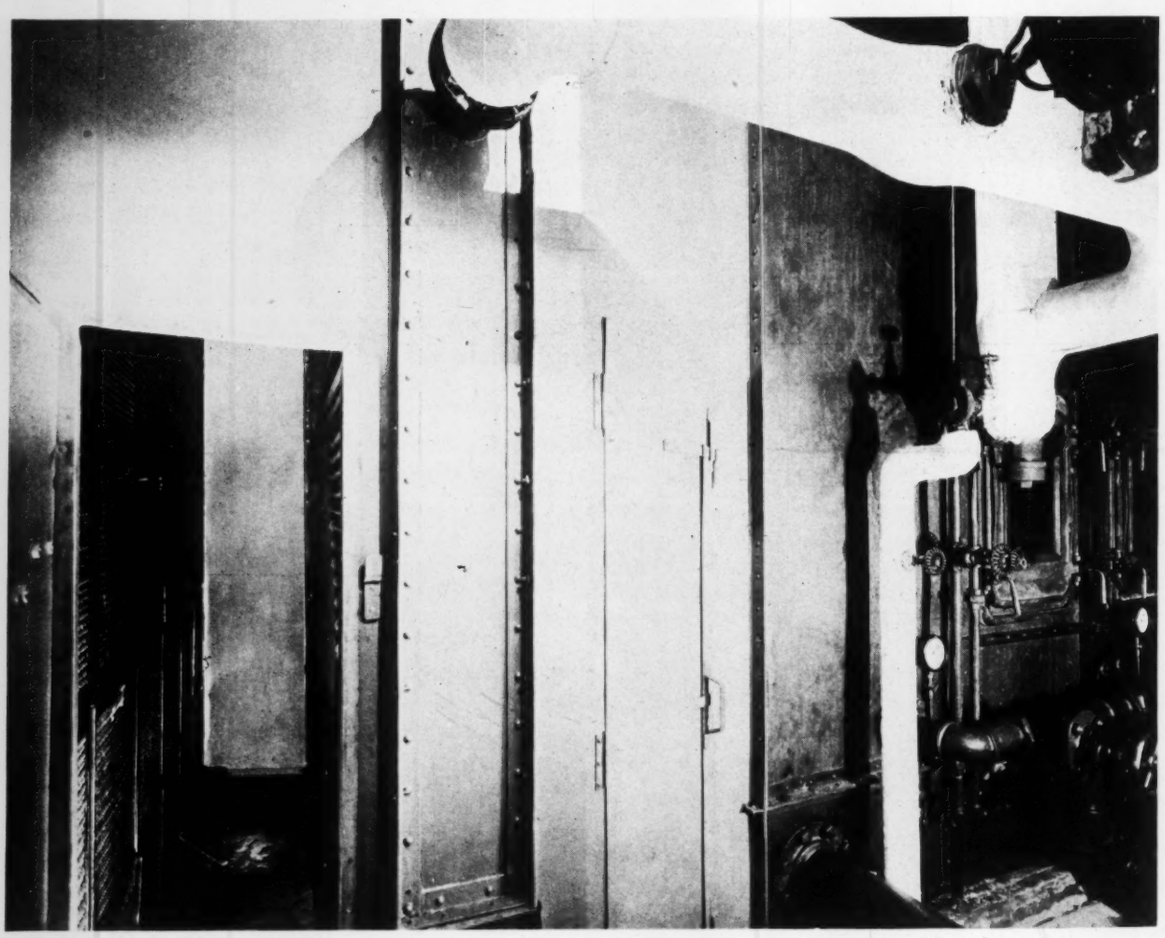


Fig. 3.—Viscous Cell-Filter Installation in Front of Air Washer. Filter is Located on Lefthand Side. Next to Filter to the Right Are Preheating Coils, Then Air Washer. Heating Coils and Fan are Further Towards the Right, But Not Visible. Air Flow from Left to Right

itself. The air is introduced through louvres, in front of which a large mesh screen should be placed. As the ordinary louvres do not insure full protection against spray droplets and snowflakes, it is advisable to place them a distance of about two to three feet from the actual filtering surface. This will allow the water and snow to settle down before it reaches the filter screen and the oil reservoir.

Where temperature differences of the summer and winter seasons are extreme, it may be necessary to use two different kinds of filter oil. In more moderate climates, and for the average application, the standard filter oils will be found to serve their purpose the year round.

Conclusions

We have seen how the performance of a filter depends a great deal upon the right amount of filtering surface, the proper layout of the filter within the system and the conditions under which the filter will have to operate. We find that it is desirable to gauge accurately the requirements of an installation in advance and to provide space for additional filtering surface, if an increase of air volume is probable in the future. Even if conditions are cramped, and unfavorable arrangements of the filter in relation to the air flow have to be tolerated, let us do the best possible under the circumstances and give the filter a fair chance.

We are sure these facts will be found worthy the careful consideration of architects, engineers or users having an interest in ventilating and air conditioning systems. If carried out as suggested, the air filter will prove an excellent investment and a permanent credit to the man who "planned it" into the system.

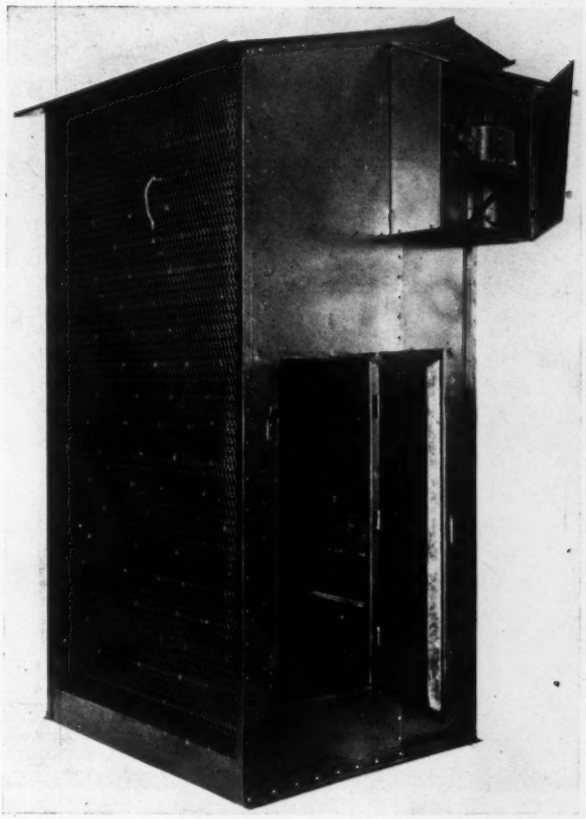


FIG. 4.—SELF CLEANING VISCOUS FILTER (MOTOR-OPERATED) EQUIPPED WITH HOUSING FOR OUTDOOR INSTALLATION. NOTE LOUVRES, PROTECTED BY SCREEN, ACCESS DOOR BETWEEN LOUVRES AND FILTER, AND SEPARATE HOUSING OVER MOTOR DRIVE. CLEAN AIR DUCT CONNECTION NOT SHOWN. AIR FLOW FROM LEFT TO RIGHT

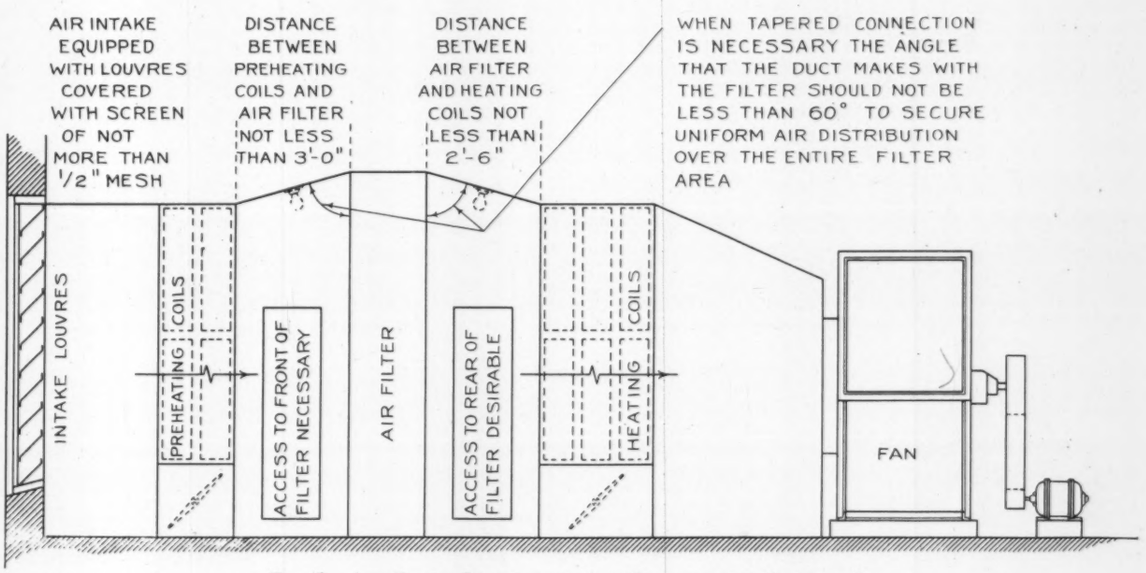

AIR INTAKE EQUIPPED WITH LOUVRES COVERED WITH SCREEN OF NOT MORE THAN 1/2" MESH
DISTANCE BETWEEN PREHEATING COILS AND AIR FILTER NOT LESS THAN 3'-0"
DISTANCE BETWEEN AIR FILTER AND HEATING COILS NOT LESS THAN 2'-6"
WHEN TAPERED CONNECTION IS NECESSARY THE ANGLE THAT THE DUCT MAKES WITH THE FILTER SHOULD NOT BE LESS THAN 60° TO SECURE UNIFORM AIR DISTRIBUTION OVER THE ENTIRE FILTER AREA
INTAKE LOUVRES
PREHEATING COILS
ACCESS TO FRONT OF FILTER NECESSARY
AIR FILTER
ACCESS TO REAR OF FILTER DESIRABLE
HEATING COILS
FAN

FIG. 5.—AIR FILTER LAYOUT FOR AIR CONDITIONING SERVICE. IF HUMIDITY CONTROL IS DESIRED, AN AIR WASHER SHOULD BE PLACED BETWEEN AIR FILTER AND HEATING COILS

Tests of Strength and Design of Welded Joints*

THE INVESTIGATION of oxy-acetylene welded pipe joints described in the following was made primarily for the purpose of determining by actual destructive pulling tests the strength of different types of joints and the most practical welded joint design. It was expected that the tests would establish the correct weld shape; amount of reinforcement and other details of joint design; the cause, extent and effect of any defects; the effect, if any, of welding or localization of heat on the pipe metal; and point out what may be expected in regard to dependability of oxy-acetylene welded joints.

Because engineers, both in this country and in Europe, have at one time or another advocated different and often individualistic joint designs for welded purposes, it was considered desirable that this investigation should include at least five types of joints, the most practical ones being selected. It was decided to enlarge the scope of the investigation to include four typical sizes, namely 4-inch, 8-inch, 12-inch and 16-inch, in the tests of these joints, because it was known to welding engineers that results seemed to vary with pipe size.

For conducting these tests, there was available the machine in the physical testing laboratory of the National Tube Company capable of gripping full size pipe sections and pulling them to destruction. The maximum size of pipe which this machine can handle is 18-in., so that this series of tests represents practically all that can be tested at the present time with available testing equipment in this country.

As there are many variables in any testing operation of this nature, it was found desirable to make six separate tests for each condition. This large number of tests made on the four typical sizes presented an opportunity for studying many problems that frequently have been presented for discussion with relation to the dependability of pipe joints and joint efficiency.

In addition to the tensile tests, compression tests were made with a few selected designs in certain sizes of pipe to observe the action of the joints under compression.

Scope of Investigation

In this investigation, which extended over a considerable length of time, 160 welded specimens were tested and compared with an equal number of plain pipe tested in the same manner. The investigation covered both lap welded (.06 to .07c) and high carbon (.30 to .40c) seamless pipe, the majority of the specimens being of the lap welded pipe.

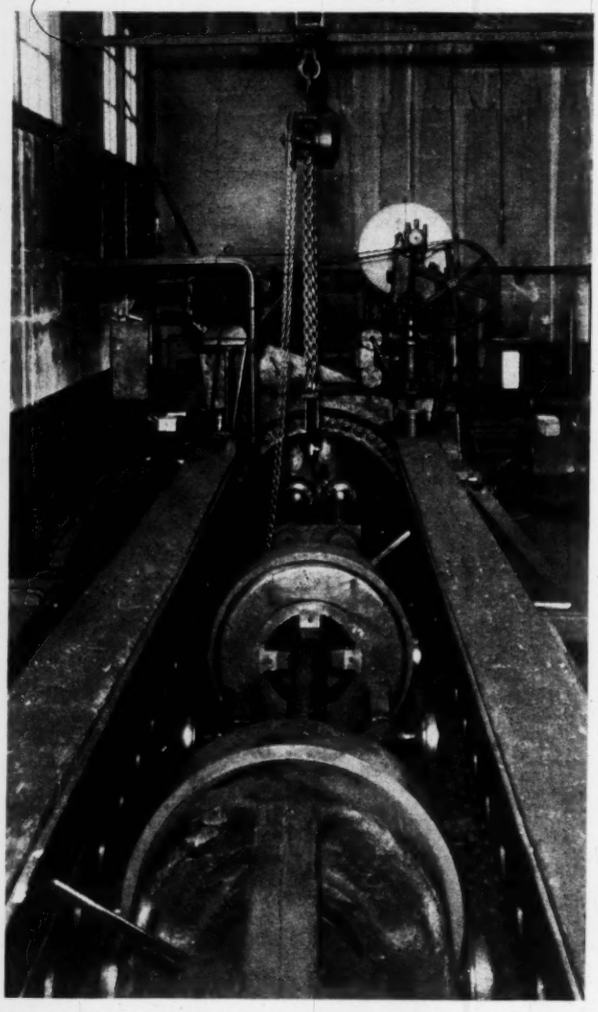


FIG. 1—600-TON HYDRAULIC TESTING MACHINE USED IN TESTING FULL SIZE PIPE SPECIMENS.

Types of Joints

The single vee type butt joint is the standard in field construction and is almost universally used for joining pipe. The preparation consists simply of machine beveling the ends of the pipe, generally at the mills, to 45 deg., making a 90 deg. included angle, when two ends are butted. This bevel does not extend completely to the inside of the pipe wall, there being an unbeveled portion at the inner side approximately 1/32 in. to 1/16 in. thick, depending upon the pipe size. This unbeveled portion of the ends assists in aligning and facilitates welding of the bottom of the vee.

The design of this joint is shown diagrammatically in Fig. 2.

* From report of sub-committee on pipe joints of the American Gas Association.

Normal Socket (B & S) Joints

This joint, termed the normal socket joint, consists essentially of a bell and spigot design in which one end of a pipe section is expanded so that the plain end of the pipe to be joined may be inserted. This joint was included in test because it is sometimes used in an attempt to avoid the effect of icicles or slight projections of weld metal inside the pipe, that sometimes occur with the butt joint. The depth of the socket is approximately 3 in., although it varies with the pipe sizes. A small space is left between the bell and the spigot. The dimensions of the socket and spacing between the bell and spigot for the different pipe sizes are shown in Fig. 3.

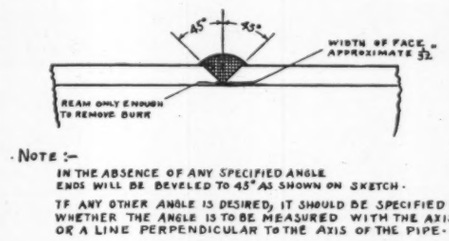

45°
32°
WIDTH OF FACE = APPROXIMATE 32
REAM ONLY ENOUGH TO REMOVE BURR
NOTE :-
IN THE ABSENCE OF ANY SPECIFIED ANGLE ENDS WILL BE BEVELED TO 45° AS SHOWN ON SKETCH.
IF ANY OTHER ANGLE IS DESIRED, IT SHOULD BE SPECIFIED WHETHER THE ANGLE IS TO BE MEASURED WITH THE AXIS OR A LINE PERPENDICULAR TO THE AXIS OF THE PIPE.

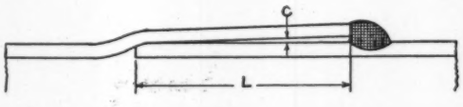
FIG. 2—SINGLE VEE TYPE BUTT JOINT

Socket Joint with Groove

This joint is essentially the same as the normal socket joint, except that a slight groove is machined in the spigot end where the weld is to be made. The purpose of this is to facilitate the welding and to determine the effect of replacing part of the pipe wall with the high strength weld metal. Design is shown in Fig. 4.

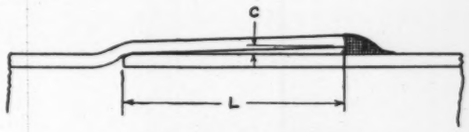
Socket with Welded Rivets

This joint is essentially the same as the normal socket joint, except that holes are punched in the bell for rivet welding. These joints were included in tests to determine the effect of welding rivets. Such designs are often specified by engineers, who question the strength of the welded joint, with the idea of increasing the strength and adding additional factors of safety. It will be noted that a crack occurred at the rivet in almost every case, and although final breaking of the specimens often occurred remote from the joints, failure of a pipe is, of course, coincident with the formation of the crack and consequent leakage. The welding of such rivets is extremely difficult and is considered by competent welding engineers to detract from the strength of the joint. Such designs should be avoided, but were added to the tests in order to actually determine the strength of such design. See Fig. 5.



SIZE	DEPTH OF SOCKET "L"	SPACE BETWEEN TUBE AND SOCKET "C"
4" O.D.	4"	5/64"
8" O.D.	4"	1/16"
12" O.D.	4"	3/32"
16" O.D.	4"	3/32"

FIG. 4—SOCKET JOINT FOR WELDING WITH GROOVE



SIZE	DEPTH OF SOCKET L	SPACE BETWEEN TUBE AND SOCKET C	DIAMETER OF HOLES FOR WELDING PLUGS	NUMBER OF HOLES FOR WELDING PLUGS
4" O.D.	3 15/16"	5/64"	5/8"	2
8" O.D.	3 15/16"	1/16"	5/8"	3
12" O.D.	3 15/16"	3/32"	7/8"	4
16" O.D.	3 15/16"	3/32"	7/8"	6

FIG. 5—SOCKET JOINT FOR WELDING WITH PINHOLES

SIZE	DEPTH OF SOCKET "L"	SPACE BETWEEN TUBE AND SOCKET "C"
4" O.D.	2 3/4"	5/64"
8" O.D.	3 1/8"	1/16"
12" O.D.	4 3/4"	3/32"
16" O.D.	5 7/8"	3/32"

FIG. 3—NORMAL SOCKET FOR WELDING

Tight Sleeve Joint

In addition to the above joints, tests were made on the tight sleeve joints shown in Fig. 6. This joint was included because it is used abroad and it was deemed desirable to determine whether such a preparation would facilitate the welding. As a matter of fact, it was found that this was the weakest and most difficult joint to make as it is impractical to fuse the metal under the flared-up lip of the bell as intended, which results in a very eccentric strain being thrown on the joint. Under no condition is this joint recommended for practical use. The same number of specimens were tested with this joint as in the socket joint with groove and the socket with welded rivets, but are not reported in detail in the test results as they represent very little practical value.

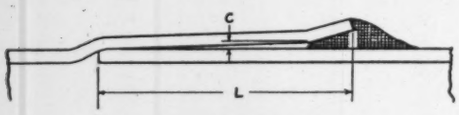
In the 4-in. and 6-in. sizes, a third of the specimens failed in the weld, and in the larger 12-in. and 16-in. sizes all but two failed in the joint at a comparatively lower load than any of the other joints. In welding, it was found that it was extremely difficult to make joints of this design even by an expert welder. Because of the difficulty in welding and low efficiency, this design is of no practical value, so the results are not presented.

Welding of Pipe Specimens

In the preparation of the test specimens, a 20-ft. length of pipe was cut into three 6-ft. sections. The remaining 2-ft. section was used for chemical analysis

and coupon test. Two of the 6-ft. sections were welded to make jointed samples of 12-ft. length, while the other 6-ft. section was used for a full size material test of the plain pipe. By this method, the effect of variation in the material was minimized in determining the joint efficiency.

During the welding, care was exercised to see that the joints were made in conformance with the best established practice. Proper control methods were



SIZE	DEPTH OF SOCKET L	SPACE BETWEEN TUBE AND SOCKET C
4" O.D.	4"	3/64"
8" O.D.	6"	1/16"
12" O.D.	6"	3/32"
16" O.D.	6"	3/32"

Fig. 6—Tight Sleeve Joint for Welding

very strictly adhered to. All the welders, as should be required in this type of work, were qualified by making satisfactory test pieces with materials similar to the pipe material and welding rod used. Most of the samples were welded by one very competent operator, but quite a few samples made by other qualified welders were included. Speed of welding was the same as would be obtainable under average conditions. No particular precautions were taken other than those ordinarily specified for making a sound and properly welded joint. It may be considered that the results obtained can be consistently expected with qualified welders when proper control methods are followed.

In the welding of the specimens, the necessary attention was paid to the penetration of the weld and the correct spacing of the pipe. No difficulty was experienced in aligning the pipe, but alignment was carefully checked so that there would be no elbow effect in the welded specimen which might influence the results.

The filler metal or welding rod used in welding all the specimens was a special, medium-low carbon, high strength welding rod.

Description of Tests

The tests were made in a 600-ton hydraulic testing machine at McKeesport, Pa. This machine, briefly, consists of heavy side plates for taking the reaction of the load, the stationary end of the machine being attached to these side plates by large bearing pins. The movable end is attached to four piston rods, which are, in turn, attached to a single hydraulic piston. The load is recorded by measuring the hydraulic pressure in the piston with mercury column gauges. Picture of this machine is shown in Fig. 1.

For holding the pipe, four tapered grips were inserted inside the ends and the pipe was gripped at corresponding points on the outside by knurled grips in the heads of the testing machine.

In the tests, the elastic limit, indicated by the falling off of the test load, the ultimate strength, total elongation, and other pertinent data were observed. For determining the cross-sectional area of the pipe, all specimens were weighed before test and the average section area computed from this weight and the length of the specimen.

Discussion of the Results

A summary of the results showing a comparison of the average welding time, strength and efficiencies of the different types of joints in various pipe sizes is given in the summary tables which appear in the full report of these tests.

These tables show the effect of pipe size on the results obtained. In the smaller sizes all the failures occurred outside the joint. In the larger sizes, although practically one hundred per cent strength was obtained, there were more failures in the joint itself. Considerable study has been given to this phase and it appears that the two important factors to be observed in securing the same maximum results in the larger sizes as is easily obtained in the smaller pipe sizes are the shape of the weld, particularly the reinforcement of the weld, and the penetration. In conducting this series of tests, the 8-in. sizes were tested first and then the 16-in. It was observed that the same relative results were not obtained and that the two factors mentioned above were important. In

Fig. 7—Typical Failure in Pipe of Normal Socket Joint. See Fig. 3

welding the 12-in. specimens, which were tested next, the welds were built up and heavily reinforced and care was exercised to secure thorough penetration. All these specimens failed in the pipe. Additional 16-in. O.D. specimens were then prepared in the same manner with equally satisfactory results. Practically all of these failed outside the joint.

Much emphasis must be placed on the shape of the weld and penetration in welding the larger size pipe. A minimum reinforcement of 25 to 30 per cent is recommended. All fillet welds should be built up well above the thickness of the bell and should be rounded outward instead of concave, so that the throat dimension is in excess of pipe wall thickness.

FIG. 8—FAILURE OF COMPRESSION TEST OF BUTT WELDED JOINT

The results show that the strongest and most efficient design, considering cost and ease of welding, is the single vee type butt joint.

Very satisfactory results were obtained with the normal socket (B&S) joint. The socket with groove joint also developed high strength which proved that the welding heat and replacement of pipe metal with weld metal had no detrimental effect. The cost of preparation is a very important factor that must be considered in adopting such designs. The joints combining socket and welded rivets were the weakest joints tested. Moreover, these are difficult to weld, costly to prepare, and are not recommended to be used in any case.

A brief study of the results would seem to indicate that all of these designs are reasonably satisfactory and all of them developed as high or higher strength than the pipe itself. This is due to the fact that sound welds made under proper control methods with good welding rod, have extremely high strength. But it should be pointed out that there are many other factors to be considered in a design in the way of even distribution of stress, the imposition of bending and eccentric stresses where the joint is strained, the cost of preparation and ease of welding. Considering all these factors, experience has shown that the single vee type butt joint is the most satisfactory and the results of these tests show it to be the strongest.

The test results of the joints in seamless (.30 to .40c) steel pipe show remarkably high strength can be obtained with the welded joint,—60,000 to 70,000 lb. per sq. in. Practically all of the joints failed to develop the strength of the pipe, but this is expected because the strength of .30 to .40 carbon pipe is higher than any commercial weld metal at present in extensive use and with which production welders are familiar. It is to be noted, however, that the strength is very high, and certainly ample for all commercial requirements.

It should be emphasized that in all these tests no failures occurred in any localized spot, such as next to the weld. Where failure occurred in the joint, it was generally in the center of the weld. This appears to establish the proof that there is no heat effect or structural weakening effect on the base metal from the welding process.

Of particular interest in these numerous tests, every one of which has been reported, is the uniformly high strength obtained in every case. This points to the dependability and strength consistency that may be expected with oxy-acetylene welded joints.

Summary and Conclusions

From the results of these numerous tests on oxy-acetylene welded joints in full size pipe, the following conclusions may be drawn:

1. A properly made oxy-acetylene welded joint is as strong as the pipe—fully 100 per cent efficient.

2. The single vee type butt joint is the most efficient and is recommended for use in welding pipe. This is in agreement with past experience as this design is practically universally used in pipe work.

3. Very high strength and nearly equally satisfactory results were obtained with the normal socket and socket with groove joints. In joint design, however, other factors must be considered, as cost of preparation, stress distribution under strain, and ease of welding, which make these designs less satisfactory than the single vee type butt joint. The weakest joint was the socket with welded rivets. This design should not be used because of cost, difficulties in welding, as well as lower joint strength.

4. Due regard must be given to penetration and to the shape of the joint, particularly the reinforcement of the weld. This is more important in the larger than in the smaller pipe sizes. Butt welds should be reinforced at least 25 per cent and should be built up so as to present a gradual increase in the reinforcement from the pipe wall to the center of the weld. Fillet welds, when used, should be built up well in excess of the wall thickness of the bell.

5. There is no detrimental heat effect or structural weakening of the pipe metal due to welding.

6. The uniformly high strength obtained in these numerous joint tests points out the dependability of properly made welds. With qualified welders under proper control methods, a pipe joint of maximum efficiency can be consistently and uniformly made by the oxy-acetylene process.

Building a New Central Heating Plant Around an Old

By C. W. Kimball, R. D. Kimball Company
Engineers, Boston, Mass.

THE original installation of central heating for the buildings which are a part of Phillips Academy in Andover, Mass., was made in 1901, and included two 60-in. dia. horizontal return tubular boilers, each 125 H.P. capacity and located in the basement of the present Chapel building. These boilers supplied steam to 8 buildings having a total of 14,500 sq. ft. of direct radiation.

The steam mains underground were laid in sectional tile conduits and they were insulated by packing the tile conduit full of ground cork coated with infusorial earth. There were approximately 1800 feet of these mains and all the returns flowed back by gravity to a return pump beside the boilers.

At the time of this installation a great many people were quite skeptical as to the success of work of this kind and the writer well remembers the many predictions of failure, of enormous heat losses and of short life that were made by those who watched the work installed.

Time and experience, however, have answered these questions and it is interesting to compare the attitude of the public with regard to this question of central heating now, and thirty years ago.

From the time of this original installation, the Academy has grown and additions and changes have been made in the central heating system to care for the buildings as they have been added. In 1904 another boiler plant was started across Main street on the west side to care for the buildings there. In 1914 the two boiler plants were combined in the west side of Main street and the plant was extended to care for Abbott Academy buildings. The additions to the boiler plant were made in the simplest way possible, as the money available for these additions was limited.

In 1928 the load on the plant had increased so much that the plant was taxed even beyond its capacity, as it was then serving 45 buildings through approximately 14,000 lineal feet of mains to 119,000 sq. ft. of direct radiation, also supplying steam for cooking, hot water, laundry, and to the engines that generated electricity for light and power. At this same time a new dining hall, library and dormitory were being built with several other buildings planned for the near future, making it necessary to provide considerable extra capacity in the central plant. As the school was developing far beyond anything planned when the original central heating system was considered, it was necessary to change the central heating plant to keep pace with the new development.

The problem from an engineering standpoint was to construct the new plant on the same site as the old plant, while the old one was in operation, without curtailing or interrupting the service in any way.

Certain parts of the old plant were also of such size and condition that it seemed wise to retain them in the new plant, rather than waste the money already invested in them.

Another feature that had to be given serious consideration was the fact that this boiler house is located in a residential district and to the northwest of the school buildings. With these points in mind, it was necessary to design every part as carefully as possible to avoid giving any annoyance from smoke, soot, coal dust or noise to these surroundings.

In determining the total boiler capacity to be installed and the size of units to be used, the records of the past were carefully examined, the additional load in sight was computed and the variation of the total load due to weather conditions was carefully checked by the past coal and weather records.

As this is primarily a heating plant with some generation of electricity, cooking and water heating, it is apparent that the peak load on the plant occurs only about 2½ months per year, 24 hours a day, and the light load a corresponding period, with an average load during the balance of the heating season.

Keeping this in mind, the size of the boiler units to meet this condition best seemed to be the 325 H.P. rated capacity. Three new boilers of this size were determined on as ample to handle the peak load with two of the existing horizontal return tubular boilers retained from the old plant, thus making a total of 1345 normal H.P. to provide steam for the total of approximately 140,000 sq. ft. of equivalent direct radiation and still provide a spare boiler at all times.

The new plan has been in operation long enough to show that the calculations as to capacity are correct, and that the new plant has removed all source of annoyance to the surrounding houses.

A reference to the plan in Fig. 1 showing the extent of the old and the new central plant building will indicate the general method of procedure that was followed in constructing the new and the size of the new building compared with the old one.

The old boilers were left in place while the new building was being built, and the three new boilers were set, piped and finally put in operation. This meant building the partition wall between the new boiler and engine room across in front of 4 old boilers, simply leaving opening through this wall through

which the old boilers could be fired. After the new boilers were in use and the old boilers were removed, these openings were bricked up and the engine room floor was finished.

As the new roof was approximately 14 ft.-0 in. above the old roof, the new roof was all completed before the old roof was removed; in fact the old roof was used as a staging from which to erect considerable of the new work.

The new boiler house was designed as plainly as possible and to harmonize with its surroundings, as it was deemed better to spend money for adequate equipment, rather than to put it into ornamentation and looks.

The building is built of brick with cast stone trimmings and with a flat tar and gravel roof. The roof is supported by steel columns which also support the boilers, coal bunkers, scales, etc.

The boiler room is 84 ft.-0 in. long by 50 ft.-0 in. wide and 31 ft.-3 in. high inside. The engine room and engineer's office extend across the front of the building occupying a space 82 ft.-0 in. long, 28 ft.-0 in. wide and 13 ft.-0 in. high. These rooms are 9 ft.-0 in. above the boiler room and below them is a pump room 22 x 28 x 9, also a toilet and locker room for the men and a room for forced draft apparatus. The plan at engine room level is shown in Fig. 2.

planned to generate to the limit of these existing units and buy the surplus current needed.

The switchboard of black slate is at one end of the engine room and is so designed that more or less of the school load can be generated or purchased as desired. To put in new generating units large enough to handle the whole of the load now being connected in the new buildings involved the expenditure of approximately $16,000 and it was decided to defer this expense until at least one year's operation as above indicated would demonstrate beyond any question of doubt the relative cost of generating or purchasing of the additional current.

The boilers selected were the cross drum type, each having 3250 sq. ft. of heating surface and set with the tubes 10 ft.-0 in. above the floor. These boilers were equipped with multiple retort stokers, using 4 retorts to each boiler. Three retorts were considered, but the operation of the units so far show that in this particular case the 4 retorts are the better and will give much more satisfactory service with less wear and replacement to the stokers and the walls.

The floor under the rear of these boilers is depressed 5 ft.-0 in. to provide for the easy removal of ashes.

The general arrangement of this may be seen by a

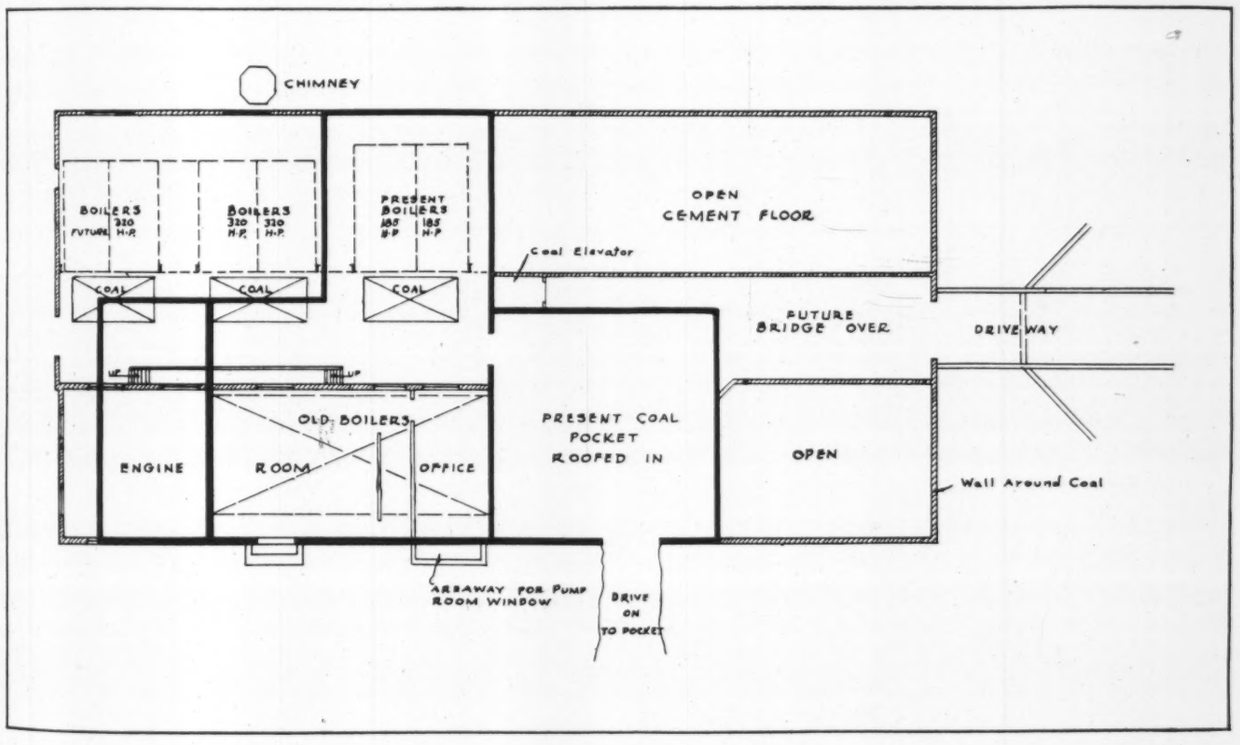


Fig. 1—Plan Showing Old and New Power House of Phillips Academy, Andover, Mass. Old Power House Is Marked by Solid Outline; New Power House by Open Outline

The engine room has a red tile floor with a wainscoting of dark green brick with a light face brick above.

In the engine room there are installed two 75 K.W. direct connected engines and 2300 volt generator, with room for additional units later. At present it is

reference to the cross section view of the boiler house in Fig. 3.

In addition to the three new boilers there are still in place and in use the two 185 H.P. horizontal return tubular boilers equipped with stokers set in dutch oven. The boilers and stokers were a part of

the old plant, but as they were only approximately 10 years old and in good condition, it was deemed wise to retain them for the light loads, spring and fall.

The steel columns and new boilers are so located that when these tubular boilers are through their usefulness, two new 325 H.P. boilers can be set in their place without disturbing the balance of the building.

Forced draft equipment has been provided for the new stokers consisting of two steam turbine driven blowers connected to the same duct and so valved that either unit may be used. Each fan is ample to provide for four boilers at 200 per cent rating.

A new chimney 150 feet high and 7 ft.-0 in. inside diameter at the top has been erected to provide adequate draft over the fire and to carry the gases well above all surrounding buildings.

All parts of the apparatus have been proportioned so that 200-250 per cent of rating may be obtained when desired without loss of efficiency, but it is expected that 150-200 per cent of rating will be the general operating range.

The coal pocket is partly covered and partly uncovered and has a capacity of 2500 tons without piling to any excessive height. The coal is now trucked from Everett, Mass., and dumped through or off the roof of the pocket. The covered pocket gives an ample supply for stormy weather with the balance used as a reserve.

There is a complete coal handling equipment consisting of an elevating and scraping conveyor equipment that lifts the coal to the roof level and conveys it along over the elevated coal bins where it drops through chutes to these bins. The conveyor has a capacity of 30 tons per hour. The elevated coal bins, three in number, each hold 20 tons. Under each bin is an automatic scale, and chutes leading to two boilers. The coal flows through the scales to the stoker hoppers and through these hoppers to the stokers as it is required by the stokers. The ashes are dumped from the end of the stokers out onto the ash tunnel floor and thence to the steam jet ash conveyor. The conveyor in turn delivers the ashes to an elevated tile ash bin with a steam heated gate to minimize the troubles due to the ashes freezing in the winter.

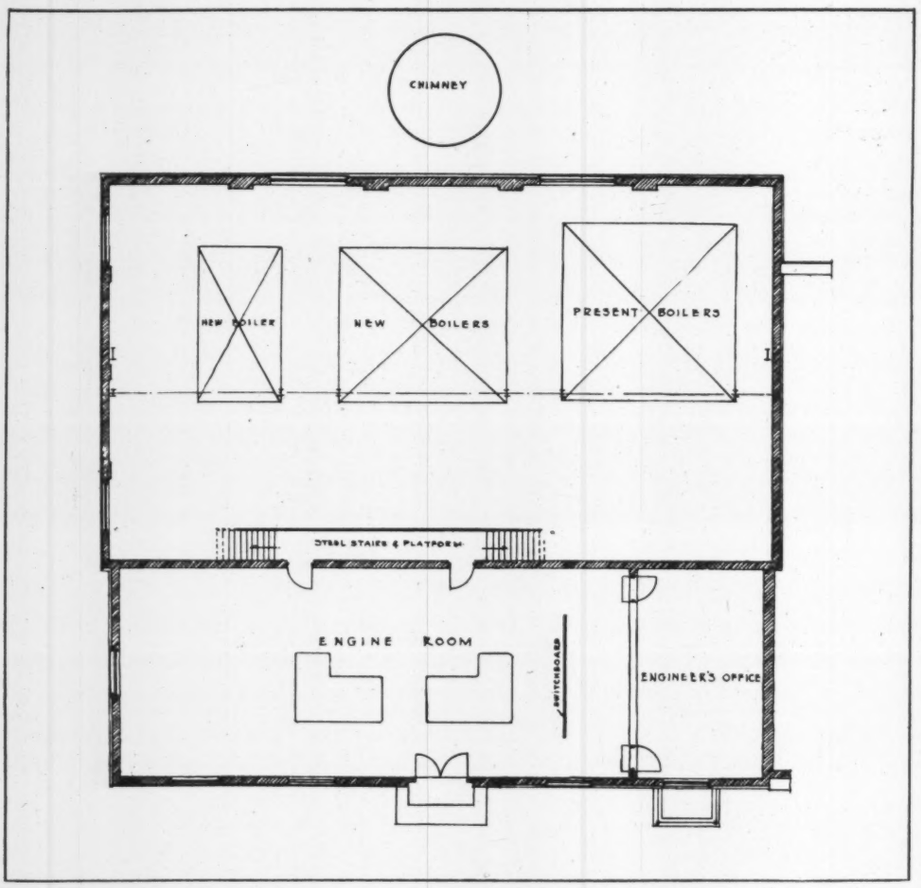


Fig. 2—Plan of Boiler House at Engine Room Level

The boilers are equipped with boiler meters and all the outgoing steam to buildings is measured by flow meters in each main line carrying steam away from the plant. There is also a flow meter on the pump discharges to measure the water pumped to the boilers with a cold water meter on the make up water.

Recorders are used to check up on the operation of the furnaces.

All the return water of condensation comes to the

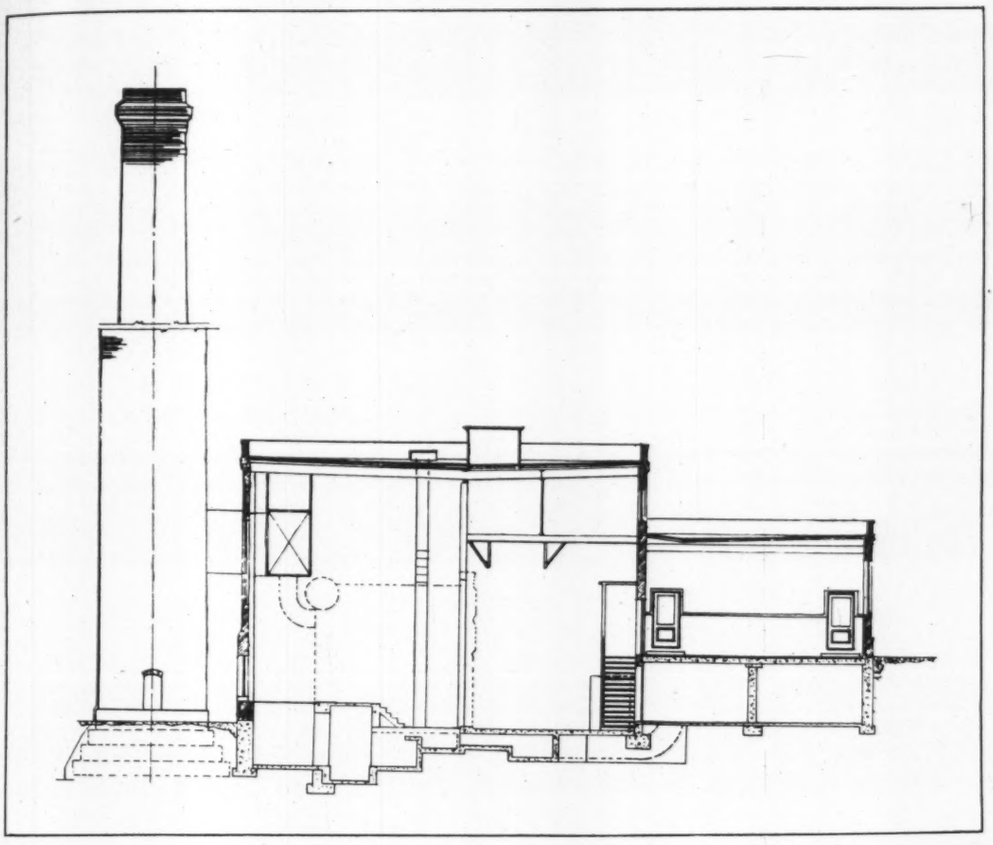


FIG. 3—CROSS-SECTION VIEW OF BOILER HOUSE

pump room where two large receivers are installed, connected so either or both receivers may be used.

There are two duplex steam driven return pumps, 7½ x 5 x 10, installed to pump the return water back to the boilers, also one small receiver and pump (6 x 4 x 6) connected to these same receivers and returns and so arranged as to operate continuously. This allows the big boiler feed pumps to care for the extra flow of returns that occurs at times on a plant of this kind. This arrangement gives a more steady and continuous flow through the meters and thus produces less fluctuation on the meter charts.

Two independent feed mains have been provided from the pumps to the boilers besides a cold water main for washing and cleaning the boilers. All the boiler feed water passes through a 500 H.P. feed water heater of closed type using exhaust from the engines and pumps for a heating medium. As the returns come back to the pump room 150 deg. to 180 deg. and the amount of make up water is small, a larger feed water heater was unnecessary.

The rebuilding of this plant was carried on through the winter and all contractors connected with the operation carried out their part of the work so carefully and well that not a minute's interruption of service occurred.

Welding Rod as Used in Making Pressure Vessels

Since a pressure vessel is a combination of plate and weld, both must be good in order to insure a high grade product. Obviously the weld metal should be at least as strong as the plate. Practically, it is desirable to have weld metal stronger than the plate, thus promoting an added factor of safety to insure against possible failure to deposit weld metal perfectly at all joints.

Certain qualifications are important in selecting welding rod for all important work. These are:

1. Ductility of weld metal.
2. Ease of manipulation of welding puddle.
3. Freedom from tendency to burn or to form laps or cold shuts.
4. Speed of deposition.

Piping for Steam Power Service

By **M. William Ehrlich**

IN THE general power field the systems are classified by their operations, such as the use of saturated steam up to 125 lb. pressure or saturated steam up to 250 lb. gage, and the plants using superheated steam.

Saturated steam is the vapor generated from the water with which it is in contact. A change in its pressure will give a change in its boiling point or corresponding temperature, and the removal of heat from saturated steam will at once start the process of condensation. However, if saturated steam be maintained at the pressure of its boiling point, and more heat added, it will become superheated, which means it will have a higher temperature than the saturated steam at the same pressure; but the removal of heat from superheated steam will not result in immediate condensation.

These characteristics of saturated and superheated steam are therefore taken advantage of in different operations or processes to gain the greatest economy from the steam generated.

The piping needed to carry such steam is governed largely by the operating pressure or by the temperature of superheat, both as to the materials and weight. A number of commercial sizes are available in standard weight and extra heavy piping suitable for these services.

Pressure Drop the Factor

The flow through this piping follows certain natural laws. Steam will pass through the system only from a higher to a lower pressure. Owing to friction, radiation, etc., the steam traveling through the pipe will be at a lower pressure at the end of the line than at the point of entry. This difference between the initial pressure and the final pressure is the pressure drop to be used as a factor in the selection of the sizes of piping required for any given condition. To obviate the necessity for extensive calculations in arriving at pipe sizes, a direct-reading chart has been prepared for a wide range of conditions, both for superheated and for saturated steam, also for standard weight as well as extra heavy pipe.

This chart, shown on the opposite page, is sufficiently close for all practical computations and aids in solving for one unknown quantity when the remaining values are known. To use this charted data, a rule, triangle or other instrument having a straight edge is required. This is used to connect the points on the several lines or scales, as follows:

1. **To Find Pipe Size:**

Connect A and B; from intersection on C connect with D; from intersection on E connect with F and at intersection on G read the pipe size.

2. **To Find Steam Flow:**

Connect F and G to intersect E; connect A and B to intersect C; then connect C and E and at intersection on D read the steam flow.

3. **To Find the Pressure Drop:**

Connect A and B; from intersection on C, connect with D; from intersection on E, connect with G, and at intersection on F read the pressure drop.

Any number of practical examples may be solved in this manner. It is only necessary to have the known or assumed values expressed in proper terms to coincide with the terms used on the scales for the various lines in the chart. A few problems have been worked out to illustrate this procedure, as follows:

Problem 1

Find the pipe size required to carry 200 pounds of saturated steam a minute with an initial pressure of 70 lb. gage and an allowable pressure drop of 0.8 lb. per 100 ft.

Answer 1

Locate the point of saturated steam on the line A of diagram, and the point of 70 lb. pressure on line B. Lay a rule across to connect these points with a straight line, and locate the point of intersection on line C of the chart. Then locate the steam flow of 200 pounds on line D and connect by rule or straight-edge with point on C, thus finding a point of intersection on line E. From this point E connect by straight line with the pressure drop of 0.8 lb. on line F and at the intersection on line G of the chart, read the answer. In this case the point of intersection on G, reads 6 in. for standard pipe and is the size required for the conditions involved.

Problem 2

What size extra heavy pipe is needed to carry 1500 lb. of steam, a minute, superheated 150 degrees? The initial pressure is 200 lb. gage and the allowable pressure drop is 1.5 lb. per 100 ft.

Answer 2

Locate 150 degrees superheat on line A and locate 200 lb. pressure on scale of line B on chart. Connect these two points by straight line and from intersection formed on line C, connect with steam flow of 1500 lb. on line D. This gives an intersection point on line E. From point E then connect with the pressure drop of 1.5 lb. on line F and read the answer at intersection on line G, which is 10-inch extra heavy pipe.

Problem 3

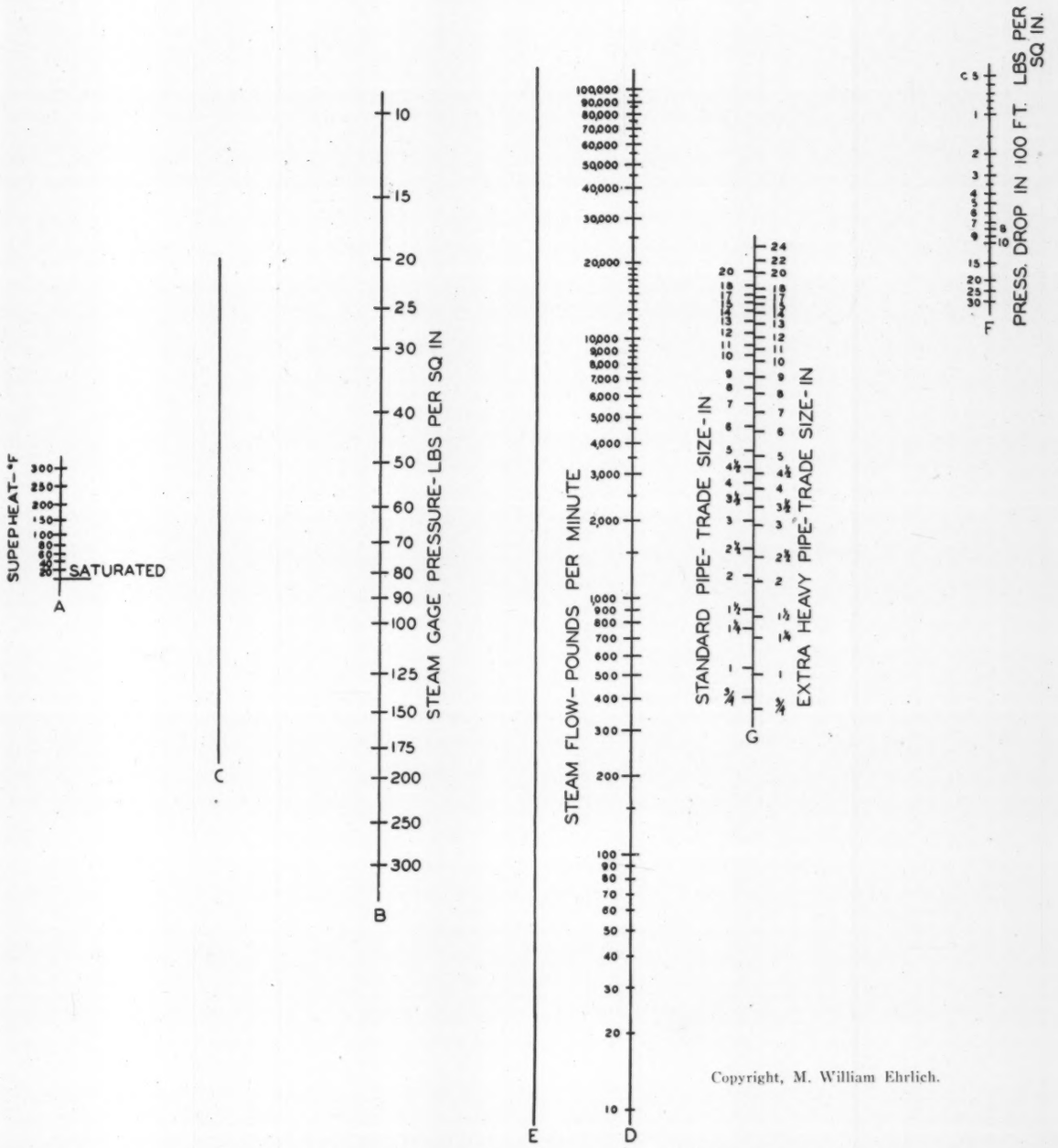
What is the flow through a 3-in. standard pipe with saturated steam at a pressure of 100 lb. gage and pressure drop of 0.5 lb. in a 100 ft.?

Answer 3

Locate 0.5 pressure drop on line F; locate 3 on line G; connect the two and find intersection on line E. Then connect the point marked saturated on line A with 100 on line B and find intersection on line C. Then lay a rule across to connect points found on C and E, and where the intersection is on line D, read the answer, which is 23 pounds of steam a minute.

Problem 4

What is the pressure drop in 100 ft. of 4-inch standard pipe when 50 lb. of saturated steam is flowing in a minute with an initial gage pressure of 50 lb.?



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Handy Chart for Determining Steam Flow and Pipe Sizes for Both Saturated and Superheated Steam. Explanation: For Pipe Sizes—Connect A and B and From C Connect with D Then from E Connect with F and Read Answer on G; For Steam Flow—Connect F and G to Intersect on E; Then Connect A and B and from C Connect with E to Intersect D and Read Answer; For Pressure Drop—Connect A and B and from C Connect with D to Intersect on E, Then Connect E and G to Intersect F and Read Answer.

Answer 4

Connect the point of saturated steam on line A with 50 on line B and find intersection on line C. From this point connect with 50 on line D to intersect with line E. Then connect from point on E with 4 on G to intersect line F. At this point read the answer, which is 0.7 lb. pressure drop per 100 ft.

From the foregoing examples illustrating the applications and uses of the data presented by the chart, it will be noted that, under the same conditions of load, length of pipe and initial pressure, a smaller size pipe will be required for a high pressure drop than for a low drop in pressure. The high pressure drop gives high velocity, while the low pressure drop gives lower velocity and larger pipe size.

When the method of pressure drop is used for pipe design, it is practically immaterial what the resultant velocity may be, so long as the pressure drop selected is suited to the conditions. Practice has shown the pressure drop in different existing piping systems to have a range from about one-half pound to five pounds per 100 ft. of pipe.

The class of service, cost of piping, etc., will be a determining factor as to permissible pressure drop.

Low Humidity Drying—III

By Malcolm C. W. Tomlinson

PSYCHROMETRIC charts that have been available were hardly satisfactory for use in reading low humidities because the relative humidity line graduations were either in 5 or 10 per cent steps. This made it necessary to estimate fractions of a per cent largely by guesswork. For this reason two new psychrometric charts were prepared to fill the need. The calculations were made very carefully and the inaccuracies of the earlier charts corrected. This will be found especially true of the wet bulb temperatures below 32 deg. fahr.

These charts were presented at the January convention (1929) of the A. S. & H. V. E. and were published in HEATING, PIPING AND AIR CONDITIONING August, 1929, on pages 316 and 317. The distinctive feature of these new charts is that the humidity lines are in steps of 1 per cent. The moisture present in a given saturated mixture can easily be calculated by means of the percentage relative humidity. Vapor pressure, total heat and moisture in grains per cubic foot lines are also drawn on the charts so that, by following a given dry bulb temperature to its intersection with any one of these three lines, it is possible to obtain, at the margin to the left of the intersection, the reading for the saturated conditions.

Of the psychrometric equipment available the only satisfactory types are those that draw the air, which is to be sampled, over the wet and dry bulbs at a rate not less than 900 feet per minute. If at all possible this rate should be 1500 to 2000 F. P. M. The work of Carrier and Lindsay has proved that the error in the wet bulb depression is due largely to radiant heat and that this error, very large in still air, decreases as the air motion is increased. In order that this error may be compensated for a set of curves are furnished in Fig. 6 which can be used when corrections are needed. It is only necessary to remember that the observed wet bulb depression, uncorrected, is not as large as it should be and that, therefore, the relative humidity figured from the observed but uncorrected readings will always be too large. The importance of taking this error into account and, if possible, building psychrometric equipment which is provided with an air movement of over 1500 F.P.M., will be realized when it is pointed out that at low relative humidities an error of only 1 per cent in the wet bulb depression may mean a reading ½ to 1 per cent higher than the actual relative humidity.

This better type of psychrometric equipment consists, therefore, of two thermometers of the mercury or vapor types (one for the wet bulb and one for the dry bulb), a suction fan, motor-driven and of the multiblade type if possible, a reservoir for clean distilled water, and a white cotton wick which fits the wet bulb snugly and thus furnishes water to the wet bulb from the reservoir. The wick should be washed clean of oil and dirt and then treated with an alkali to remove all traces of acid. Each day a clean wick should replace the one in use the previous day as salts from the water and dust clog the cotton pores and thus retard capillary action on the part of the water. An illustration of this psychrometer is not furnished as the construction is very simple. Recording psychrometers of the type described can be purchased. They have a number of disadvantages. The charts are small and it is impossible to read the temperatures to ¼ of a degree. Furthermore they are usually supplied with rather expensive wicks which are hard to clean.

Measuring Relative Humidities Under 10 Per Cent

When relative humidities much lower than 10 per cent must be measured the rate of evaporation is so high that the psychrometric equipment just described is not dependable. A method which assures considerable accuracy but which has never been developed beyond the indicating stage measures the dewpoint of the air sampled. This method is based on the fact that, when the dry bulb temperature of an air-water vapor mixture is reduced sufficiently, a temperature is reached where the moisture in the mixture is exactly equal to the moisture which is necessary for a saturated solution, or mixture, at the latter temperature. This temperature is known as the dewpoint of the original air-water vapor mixture because dew, or frost in case the dewpoint is below

Table 2

VAPOR PRESSURES FOR SATURATED AIR-WATER VAPOR MIXTURES

DEG. F.	MMHG.	DEG. F.	MMHG.	DEG. F.	MMHG.
−40	.0966	14	1.950	68	17.535
−38	.1094	16	2.149	70	18.765
−36	.1237	18	2.367	72	20.070
−34	.1398	20	2.603	74	21.453
−32	.1577	22	2.862	76	22.922
−30	.1796	24	3.171	78	24.617
−28	.2021	26	3.480	80	26.271
−262271	.2271	28	3.816	82	28.021
−24	.2549	30	4.182	84	29.870
−22	.2859	32	4.579	86	31.824
−20	.320	34	4.962	88	33.888
−18	.359	36	5.370	90	36.068
−16	.401	38	5.807	92	38.369
−14	.448	40	6.274	94	40.796
−12	.505	42	6.822	96	43.595
−10	.563	44	7.360	98	46.302
− 8	.627	46	7.936	100	49.157
− 6	.698	48	8.551	102	52.160
− 4	.776	50	9.209	104	55.324
− 2	.862	52	9.910	106	58.65
0	.956	54	10.658	108	62.14
2	1.060	56	11.453	110	65.82
4	1.175	58	12.302	112	69.69
6	1.312	60	13.290	114	74.12
8	1.450	62	14.256	116	78.40
10	1.602	64	15.284	118	82.87
12	1.769	66	16.374	120	87.58

freezing, will appear on adjacent metal or glass surfaces. This dew is a portion of the water vapor, heretofore present in the air-water vapor mixture, which has been squeezed out by the reduction of the temperature in the same manner that rain and snow are created. The device consists, primarily, of a thin sheet of platinum coated with platinum black. The enclosure above the platinum surface houses a source of light, a magnifying glass, and means for introducing and removing air. Underneath the platinum a means of refrigeration is provided so that the upper surface of the platinum may be cooled to the unknown dewpoint. Provision for measuring the temperature of the dewpoint may be made by attaching the hot junction of a thermocouple to the upper surface of the platinum plate. At the instant dew or frost appears the temperature of the platinum black surface and the temperature of the entering air should be noted. From Table 2 it will then be possible to obtain the vapor pressures, at saturation, for these two temperatures. By dividing the vapor pressure of the dewpoint by the vapor pressure of the entering air the relative humidity can be obtained. The vapor pressures given in the table are from the International Critical Tables and those for temperatures below freezing are for over ice. These vapor pressures are, of course, for saturation.

It is well to bear in mind that, unless proper precautions are taken to prevent it, the relative humidity of room air conditioned below 5 per cent will fluctuate considerably. Therefore it is likely to be rather difficult, by the indicating method just described, to obtain two readings in succession which will be identical. This is not always necessary, for an average idea of the humidity may be sufficient and, under such conditions, the indicating method is perfectly satisfactory.

Recording Low Humidities

A method for recording low relative humidities has been devised which is very suitable for laboratory purposes. This consists of the special development of a thermal conductivity method for gas analysis first developed by the Bureau of Standards. The air-water vapor mixture to be analyzed is passed over an electrically heated platinum wire and, by means of a Wheatstone bridge, the resistance of this wire is compared with the resistance of a similar electrically heated platinum wire over which dry air is passed. The temperature of the two wires is kept constant by an oil bath and a wet battery furnishes a constant e. m. f. Finally an electrical recorder registers the relation between the dried air and the air under test in terms of milliamperes. A calibration chart interprets the results in terms of relative humidity for a given temperature. Calibration of the recorder is had by means of gravimetric analysis in which a definite volume of air is passed through phosphorus pentoxide for a definite length of time. Original and final weights of the drying tubes furnish the actual moisture removed from the air. It is possible to remove practically all of the moisture in the air sampled. Relative humidity can then be calculated.

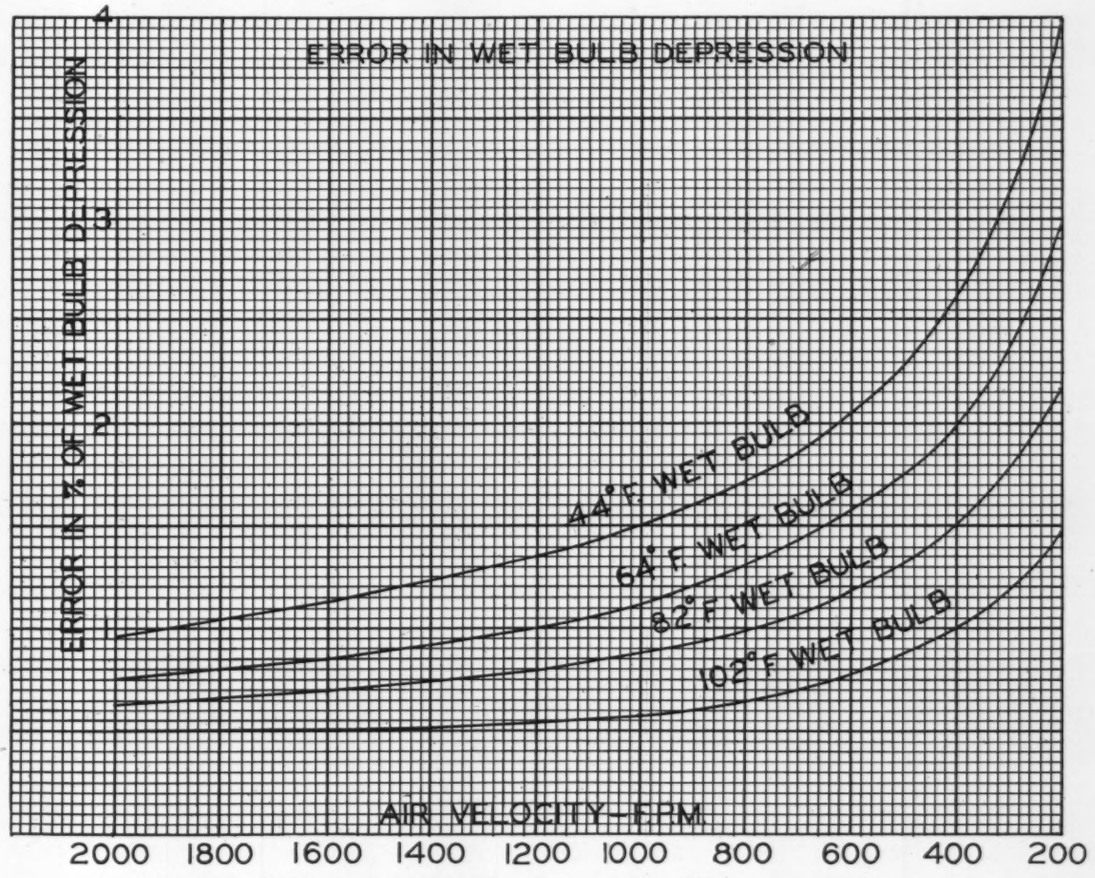


Fig. 6. Chart for Determining Error in Wet Bulb Depression.

Design of Flanged Pipe Joints—II

By Sabin Crocker

THE principal stresses in a pipe flange are radial and hoop stresses which are produced through bending induced by tension in the bolts acting through a lever arm whose length is the overhang of the bolts beyond the contact surface. Neglecting expansion strains, these stresses are about as severe when the bolts are initially tightened as when pressure is on the line.

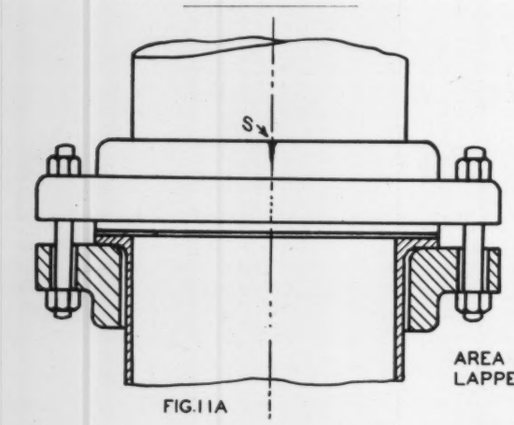
Since stress in the flange depends on bolt tension, it is convenient to evaluate it in relation to bolt tension which has previously been designated as t. The physical properties of bolt material usually are superior to those of the flange in which the bolt is used. Hence the flange should be designed for a lower working stress than is permissible in the bolt. If s represents maximum stress in the flange in lb. per sq. in., this relation may be expressed as $s < t$ or $s = ct$ where c is a stress ratio factor less than unity.

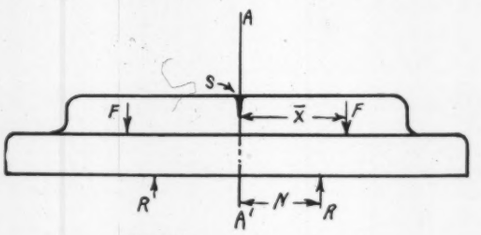
Another reason for seeking to keep c less than unity is to avoid the possibility of cracking or permanently dishing a flange by overstressing the bolts during erection. Most engineers prefer to have an overstressed bolt snap while being tightened rather than to run the risk of ruining a flange, fitting or valve, since the delay and cost of replacement is comparatively negligible in the case of the bolt. While the tension set up in the bolt through wrench torque might be only that required from theoretical considerations such as those explained in connection with Tables I and II in Part 1 of this article, published in the August issue, it actually may be anything up to the yield point of the bolt material unless some adequate system of control is set up to regulate the actions of the erection crew.

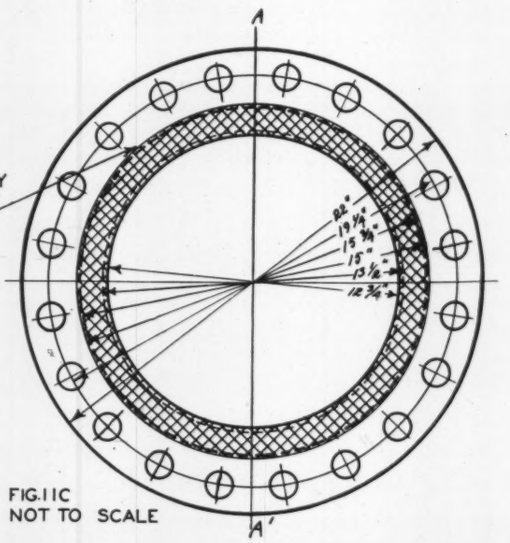
Computing Stresses in Pipe Flanges

Several more or less exact methods have been advanced for computing stress in pipe flanges. The outstanding reference on this subject is the paper by Everett O. Waters and J. Hall Taylor on "The Strength of Pipe Flanges," presented at the A. S. M. E. spring meeting in 1927 and published in the mid-May 1927 issue of Mechanical Engineering. This paper evoked considerable discussion from various engineers, regarding both the solution offered in the paper and alternative methods of dealing with the problem. The discussion was published in the December 1927 issue of Mechanical Engineering. The method described by Waters and Taylor treats both flat rings and hubbed flanges by the St. Venant theory of maximum strain so as to obtain the combined stress due to the two principal stresses which in this case are hoop tension and shear. St. Venant assumed that the criterion for the failure of the material was for the maximum strain to exceed a certain limit. The maximum strain is a function of the two principal stresses, Poisson's ratio, and the modulus of elasticity of the material. Certain

FIG. 11—MAXIMUM STRESS IN SECTION A-A', 12-IN. 600 LB. AMERICAN STANDARD VAN STONE FLANGE







limitations exist in the application of the Waters-Taylor formulas in that they are not directly applicable to fitting flanges and the physical significance of the formulas is hard to visualize and understand. Hence, there are circumstances under which it is convenient to resort to the more easily applied methods based on the Rankine criterion for failure.

The Rankine Criterion for Failure

Rankine assumed that the criterion for the failure of a material was for the maximum normal stress to exceed a certain limit. In other words, if more than one stress, such as hoop stress or radial stress, exists, the larger of these two stresses is the determining factor. It has been rather general practice in the past to calculate the strength of any structural or machine part with reference to the weakest section, by passing a plane through this section normal to the direction of the principal stress and determining the maximum intensity of stress per unit area.

The application of both the St. Venant and the Rankine criterions of failure as applied to pipe flanges are explained in subsequent paragraphs.

METHOD FOR COMPUTING MAXIMUM FIBRE STRESS IN SECTION A-A′ OF VAN STONE FLANGE WITH HUB, USING RANKINE CRITERION FOR FAILURE

The problem is to determine the maximum fibre stress in the flange due to initial setting up of the bolts:

In order to demonstrate the method used, computations are given below for an American Standard 600 lb. steel Van Stone 12 inch flange, having a 2 inch hub. (See Fig. 11).

The flange is subjected to forces acting through the bolts and to resultant forces acting along the ring portion in compression against the lapped over end of the pipe. The flange may be considered as a beam acted upon by these forces.

Consider the stress at the section A-A′, Fig. 11B. The bolt force in each half of the flange can be resolved into a single force F; and the resultant forces acting against the bottom of the flange can be resolved into a single force R, which is equal to F.

The bolt forces are considered as acting uniformly along the circumference of the bolt circle. The force F then acts at the center of gravity of the semi-circumference, which is at a distance x from the section A-A′.

$$\bar{x} = \frac{d}{\pi} = \frac{19\frac{1}{4}}{\pi} = 6.12 \text{ in.}$$

The resultant forces are considered as acting uniformly on the ring area in contact with the lapped over end of the pipe, shown in Fig. 11C. The force R then acts at the center of gravity of the half ring, which is at a distance N from the section A-A′

$$N = \frac{2(d_0^3 - d_1^3)}{3\pi(d_0^2 - d_1^2)} = \frac{2[(15)^3 - (13.5)^3]}{3\pi[(15)^2 - (13.5)^2]} = 4.52\text{-in.}$$

Let t = the initial tension in the bolts, lb. per sq. in., at root of threads. There are ten 1¼-in. bolts in each half of the flange, each bolt having an effective area at root of threads of 0.930 sq. in. (See Table V in Part 1, August issue.)

$F = 10 \times 0.930t = 9.30t.$

Let M be bending moment acting on the section A-A′.

$M = 9.30t\ (6.12 - 4.52) = 14.88t$ lb. in.

In order to compute the stress it is necessary to know the position of the neutral axis and the amount of inertia of the section A-A′.

The position of the neutral axis, NN, above the axis XX, Fig. 11D, is found by taking the area moments about XX and dividing the summation of these area moments by the summation of the areas.

$$-\frac{\Sigma Ay}{\Sigma A} = \frac{[22 - 12.75 - (2\times 1\frac{3}{8})]\ 2\frac{5}{8}\times 1\frac{5}{16} + (15\frac{3}{4} - 12\frac{3}{4})\times 2\times 3\frac{5}{8}}{[22 - 12.75 - (2\times 1\frac{3}{8})]\ 2\frac{5}{8} + (15\frac{3}{4} - 12\frac{3}{4})\times 2} = 1.92\text{-in.}$$

(6.50) (3)

The moment of inertia of the section is found by the parallel axis theorem, I_0 being the moment of inertia of an area about its axis and y being the distance between this axis and the neutral axis of the entire section.

$$I_{NN} = I_0 + Ay^2$$

$$= \frac{6.50 \times (2\frac{5}{8})^3}{12} + (6.50 \times 2\frac{5}{8})\ (1.92 - 1\frac{5}{16})^2 = 16.17\text{-in.}^4$$

$$= \frac{3 \times (2)^3}{12} + (3 \times 2)\ (3\frac{5}{8} - 1.92)^2 = 19.48\text{-in.}^4$$

Total moment of inertia about neutral axis = 35.65-in.^{4}

The maximum stress on the section occurs at the top of the hub, and if this stress is sufficient, failure will occur as shown in Fig. 11 at S.

Let S = the stress at this point

$$S = \frac{My}{I} = \frac{14.88t\ (4\frac{5}{8} - 1.92)}{35.65} = 1.13t$$

The stress ratio of 1.13 for a 12-in. 600 lb. Van Stone flange with 2-in. hub as arrived at above is not in agreement with what was previously stated regarding the desirability of having the stress ratio between flange and bolt less than unity. The conditions of loading as assumed above are considerably more severe than those probably obtaining in actual steel flanges which are capable of deflecting a limited amount without taking a permanent set. A more nearly true average condition of loading for a steel flange is represented by considering the bolt forces as acting on the inner edge of the bolt holes rather than on the bolt circle, because as deflection takes place, due to initial screwing up of the nuts, the load is no longer uniformly distributed on the under side of the nuts. This shifting of the points of application of the bolt forces is clearly born out by tests in which the imprints of the nuts on the flange are as indicated in Fig. 12.

If, in the above example, the bolt forces are considered as acting at the inner edge of the bolt holes, the stress ratio is reduced to 0.83 (meaning that the maximum stress in flange material is 83 per cent of the bolt tension at root of threads) and the requirement of having the maximum flange stress less than the bolt tension is satisfied. Generally speaking, with reference to the assumption of bolt forces acting at the inner edge of the bolt holes rather than on the bolt circle, the reduction in computed flange stress for the American Steel Flange Standards is of the order of 25 per cent.

With some types of joints such as the raised face with

gasket, there is a tendency for the gasket to be pinched more tightly at its outer circumference than at the bore of the pipe, due to the position of the bolts beyond the raised face and the deflection of the flanges under bolt loading. This tendency is borne out clearly by asbestos gaskets which have been removed from joints which have been made up. Such gaskets become wedge-shaped, being very thin at the outer edge and comparatively thick at the pipe bore. This action indicates a further tendency to relieve bending stress in the flange, since it constitutes a shortening of the moment arm of the bolts by moving the gasket reaction out from the median line of the gasket to a position toward its outer edge.

METHOD FOR COMPUTING MAXIMUM FIBRE STRESS IN SECTION A-A′ OF A LOOSE FLANGE WITHOUT HUB, USING RANKINE CRITERION FOR FAILURE (See Fig. 12)

Using the same notation and method as in the case of the loose flange with hub, and employing the additional symbols given in connection with Fig. 12, the equation for maximum stress in Section A-A′ of a Van Stone flange without hub, can be written:

$$s = \frac{My}{I} = \frac{at/2 \times L \times T/2}{bT^3/12} = \frac{3aLt}{bT^2}$$

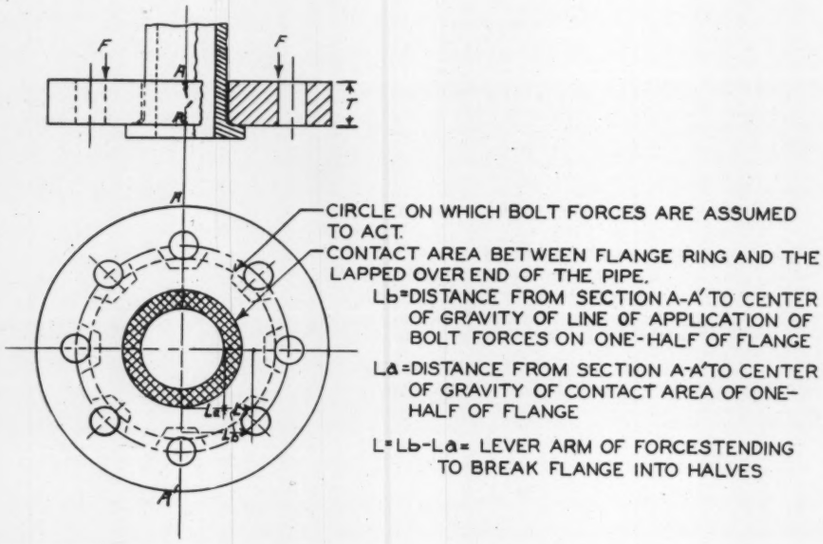

F
A
A′
T
CIRCLE ON WHICH BOLT FORCES ARE ASSUMED TO ACT.
CONTACT AREA BETWEEN FLANGE RING AND THE LAPPED OVER END OF THE PIPE.
Lb=DISTANCE FROM SECTION A-A′ TO CENTER OF GRAVITY OF LINE OF APPLICATION OF BOLT FORCES ON ONE-HALF OF FLANGE
La=DISTANCE FROM SECTION A-A′ TO CENTER OF GRAVITY OF CONTACT AREA OF ONE-HALF OF FLANGE
L=Lb-La= LEVER ARM OF FORCES TENDING TO BREAK FLANGE INTO HALVES

FIG. 12—MAXIMUM STRESS IN SECTION A-A′ OF VAN STONE FLANGE WITHOUT HUB

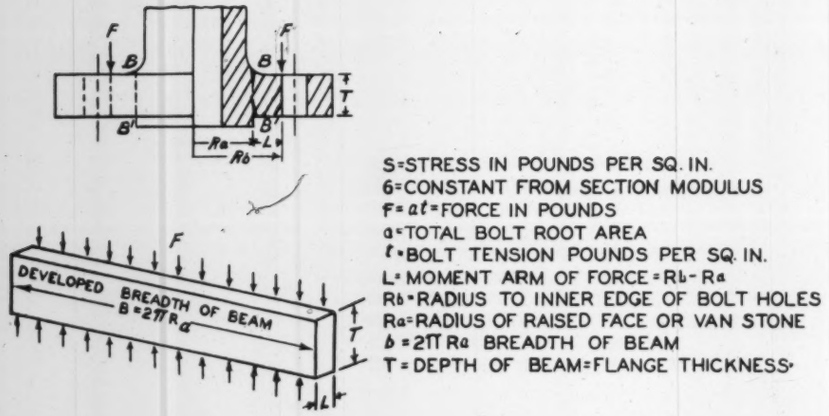


FIG. 13—MAXIMUM STRESS IN RING SECTION B-B′ OF RAISED FACE FITTING AND VAN STONE FLANGE

In this case the bolt forces are again considered as acting at the inner edge of the bolt holes, for the reason explained in the preceding paragraph.

METHOD FOR COMPUTING MAXIMUM FIBRE STRESS IN SECTION B-B′ OF RAISED FACE FITTING AND LOOSE FLANGES, USING RANKINE CRITERION FOR FAILURE (See Fig. 13)

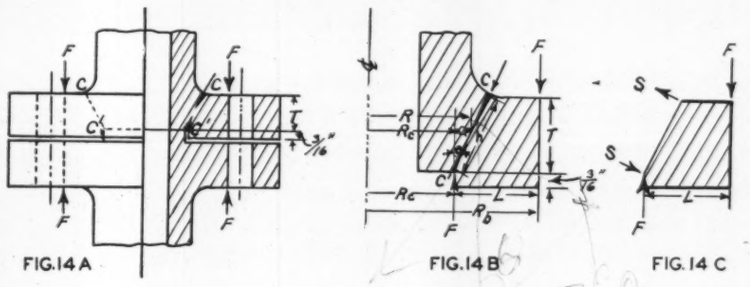
In a fitting flange with raised face the weakest section is evidently as shown at B-B′ in Fig. 13. Using the previous notation and employing the addition symbols show in Fig. 13, the equation for this case may be written:

$$s = \frac{6FL}{bT^2} = \frac{6aLt}{bT^2}$$

This same section exists in hubbed loose flanges directly above the outer edge of the lapped-over pipe end. In checking the strength of a Van Stone flange by the above methods it is necessary to consider stress in section B-B′ as well as in the section A-A′ previously described. The bolt forces have again been assumed as acting at the inner edge of the bolt holes.

METHOD FOR COMPUTING MAXIMUM FIBRE STRESS IN SECTION C-C′ OF "SMALL MALE-FEMALE" AND SIMILAR TYPES OF FITTINGS, USING THE RANKINE CRITERION FOR FAILURE (See Fig. 14)

The formula developed below for calculating the maximum stress obtaining in diagonal ring section C-C′ is particularly applicable to the small male-female fittings,



but can be applied equally well to any fittings which are suspected of being weak in this section.

FIG. 14—MAXIMUM STRESS IN DIAGONAL RING SECTION C-C′ OF FLANGED FITTING SMALL MALE-FEMALE TYPE JOINT

The symbols used in the development of the formula are: (See Fig. 14)

s=stress in pounds per square inch

F=a t=total bolt load in pounds

a=total bolt area at root of threads, square inches

t=bolt tension, pounds per square inch

$L=R_b-R_c$=moment arm of external force couple composed of bolt forces and gasket reactions (the bolt forces have again been assumed as acting at the inner edge of the bolt holes.)

R_b=radius of circle tangent to inner edge of bolt holes

R_c=outer radius of small female

$R=R_c+Q$=radius of mean circle

$Q=\sin\theta\, h/2$ (See Fig. 14)

$h=\dfrac{T}{\cos\theta}$=depth of section C-C′

θ=angle between center line of fitting and slope of section C-C′. (This angle may be obtained by laying out flange to scale and assuming a fillet radius.)

T=flange thickness minus depth of female in the case of the small male-female fittings.

From the above relations it follows that the total bolt load acting upon the flange is

$$F=a\,t$$

The bolt forces and resisting stresses may be found per unit length of the circumference of the mean circle of radius R by dividing the total forces and stresses by $2\pi R$.

The external moment of the couple composed of the bolt forces and the equal and opposite gasket reaction per unit length of the mean circumference is:

$$M=\frac{F\,L}{2\pi r}=\frac{a\,t\,L}{2\pi r}$$

The moment of resistance of the rectangular section of depth h per unit length of the mean circumference is the product of the section modulus and the maximum stress in the outer fibres of the flange material:

$$M=s\frac{I}{c}=\frac{s\,b\,h^2}{6}$$ (See Marks Handbook, Second Edition, Page 416)

Substituting the value of $h=\dfrac{T}{\cos\theta}$

$$M=\frac{b\,t^2}{6\cos^2\theta}$$

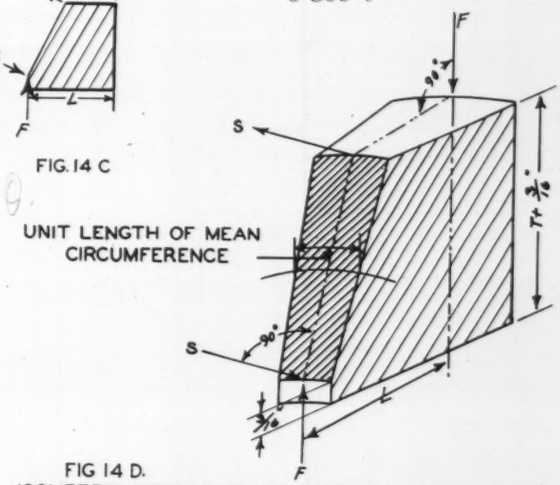


FIG 14 D.
ISOMETRIC VIEW OF UNIT LENGTH OF SECTION C-C′ HEAVILY SHADED AREA REPRESENTS PORTION OF DIAGONAL RING SECTION C-C′ OR THE CRITICAL SECTION OF THE SMALL FEMALE FITTING FLANGE.

Equating the external and internal moments and noting that for a unit length of the mean circumference b, the breadth of the section, is equal to unity, the following equations result:

$$\frac{s\,d^2}{6\cos^2\theta}=\frac{a\,t\,L}{2\pi R}$$

$$\text{or } s=\frac{6\cos^2\theta\,a\,t\,L}{2\pi R\,T^2}$$

$$s=0.96\cos^2\theta.\frac{a\,L}{R\,T^2}.t$$

Editor's Note: Part III of "The Design of Flanged Joints" will finish this study. It will appear in October.

Pontchartrain Club

The new home of this Detroit athletic club is under construction and will be ready in February. An elaborate air conditioning installation is part of the mechanical equipment.

The ventilating system will pump 3,863,400 cubic feet of washed air per hour, while the refrigerating system used for cooling has a capacity of 400,000 pounds of ice per day. In the swimming pools, ultra-violet ray machines will be used to give the benefits of sunshine.

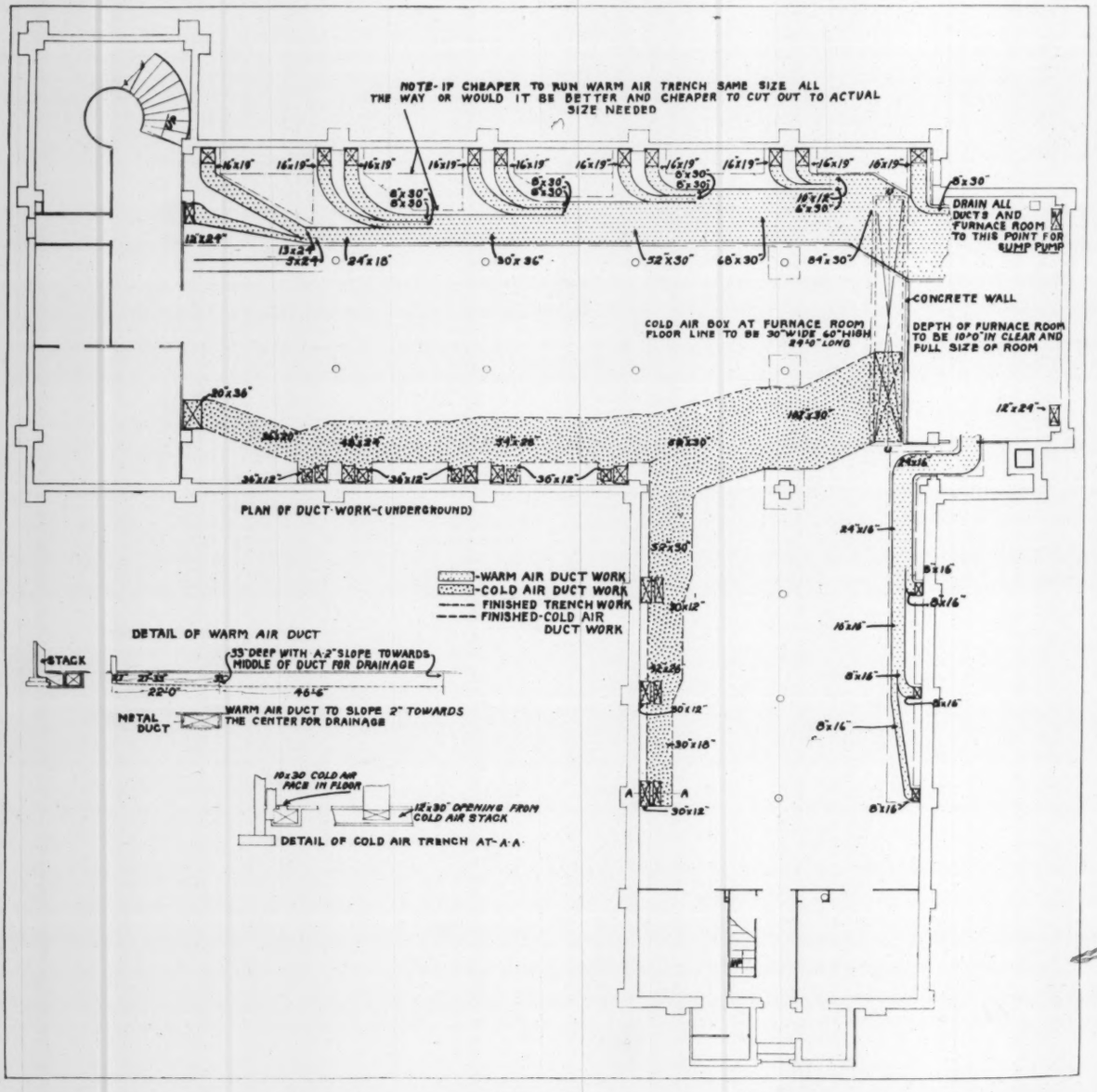
The clubhouse, which will stand fifteen stories high, will cost $4,000,000.

Computing a Warm Air System

By E. B. Langenberg

THE problem of heating churches with a warm air system, either gravity or mechanical, requires fully as much careful consideration and accuracy as when any other type of heating plant is used. Too much of this class of work, however, has been done by guess, and it has only been due to the inherent ability of a warm air furnace to produce excess capacity by overfiring that any degree of satisfaction has been recorded by the occupants of buildings wherein the heating plant has been improperly designed and installed.

With the advent of the research work done by the experiment station of the University of Illinois in co-operation with the National Warm Air Heating Association, a greater degree of accuracy is developing among the installers and much better engineering principles are being utilized to perfect the design of church installations. The same might also be said for all classes of warm air heating. The standard code, evolved from the laboratory and years of practice, fully covers the installation of this type of system in residential work. This code, however, is not very well adapted to large buildings and for this reason the installer has to turn to the engineer


NOTE-IF CHEAPER TO RUN WARM AIR TRENCH SAME SIZE ALL THE WAY OR WOULD IT BE BETTER AND CHEAPER TO CUT OUT TO ACTUAL SIZE NEEDED
DRAIN ALL DUCTS AND FURNACE ROOM TO THIS POINT FOR SUMP PUMP
CONCRETE WALL
COLD AIR BOX AT FURNACE ROOM FLOOR LINE TO BE 30" WIDE 60" HIGH 24'-0" LONG
DEPTH OF FURNACE ROOM TO BE 10'-0" IN CLEAR AND FULL SIZE OF ROOM
PLAN OF DUCT WORK-(UNDERGROUND)
WARM AIR DUCT WORK
COLD AIR DUCT WORK
FINISHED TRENCH WORK
FINISHED-COLD AIR DUCT WORK
DETAIL OF WARM AIR DUCT
STACK
33" DEEP WITH A 2" SLOPE TOWARDS MIDDLE OF DUCT FOR DRAINAGE
METAL DUCT
WARM AIR DUCT TO SLOPE 2" TOWARDS THE CENTER FOR DRAINAGE
10x30 COLD AIR FACE IN FLOOR
12x30" OPENING FROM COLD AIR STACK
DETAIL OF COLD AIR TRENCH AT-A-A

for more accurate methods of computing heat losses and better methods of installations.

An effort will be made in this article to show how computations are made in this field, together with the application of the principles necessary to provide comfort and good air conditions to the occupants of a church.

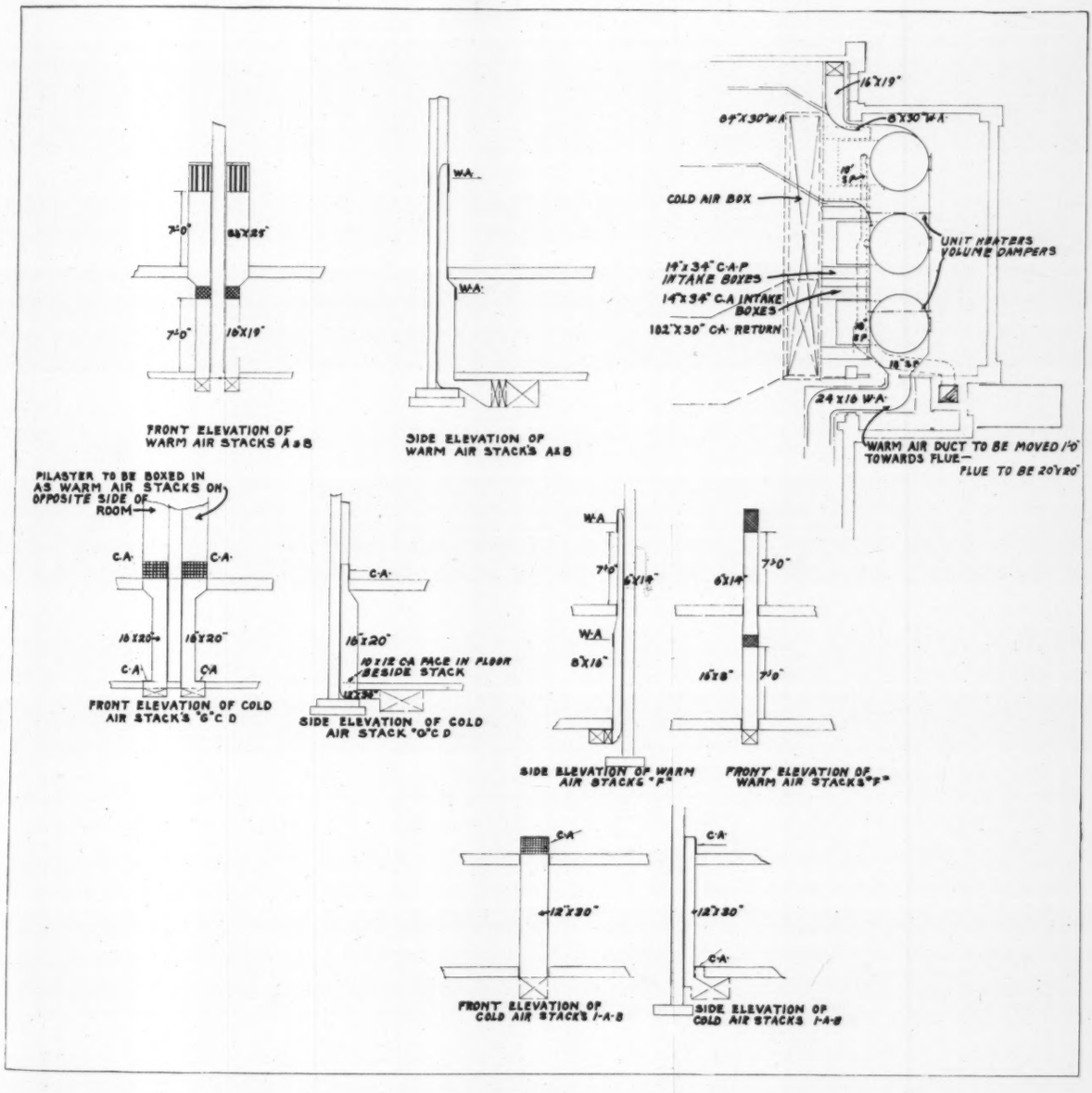
A Church Job

The church used for this illustration is of Gothic design, the main auditorium being 36 × 106 × 55 and the wing being 31 × 61 × 14. A 9-ft. finished basement is built under each wing for assembly purposes. The heater room is directly under the altar. The depth of this heater room is 14 feet, thus making it possible to run both warm air and cold air pipes underneath the basement floor.

The most difficult problem from the heating contractor's viewpoint was the fact that the building was built on solid limestone and the space for the air ducts had to be excavated in solid rock beneath the basement floor. In the case of the warm air pipes it was necessary to make the excavation larger than the pipes as under no consideration were these metal ducts permitted to touch any of the rock work. The reason for this is that a heavy heat loss occurs by conduction when in contact with stone or other material. These pipes were covered with air-cell paper. Some pitch to the pipes was possible, although the pitch was not sufficient for gravity flow of air. Manholes were provided at several points for access to this work.

In the case of the cold air excavation, the bottom and sides were made smooth with cement and it was not deemed necessary to use metal pipes as the heat loss in these ducts would not justify the expense for them.

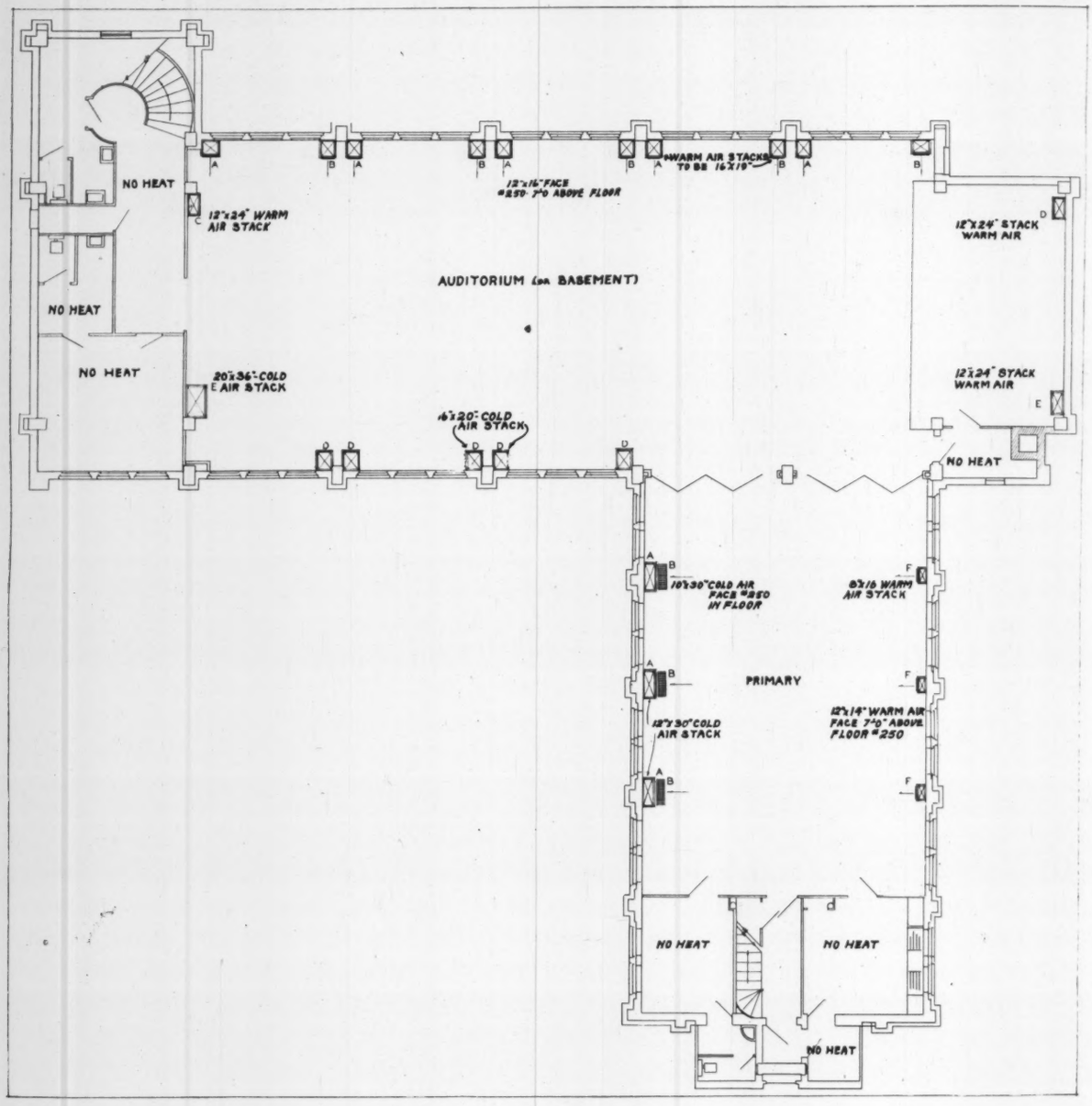
The entire underground system had a natural


FRONT ELEVATION OF WARM AIR STACKS A & B
SIDE ELEVATION OF WARM AIR STACKS A & B
COLD AIR BOX
UNIT HEATERS VOLUME DAMPERS
19"x34" C.A.P. INTAKE BOXES
14"x34" C.A. INTAKE BOXES
18"x30" C.A. RETURN
24"x18" W.A.
WARM AIR DUCT TO BE MOVED 1'0" TOWARDS FLUE
FLUE TO BE 20"x20"
PILASTER TO BE BOXED IN AS WARM AIR STACKS ON OPPOSITE SIDE OF ROOM
FRONT ELEVATION OF COLD AIR STACKS "G" C D
SIDE ELEVATION OF COLD AIR STACK "D" C D
16"x20"
10"x12" CA PACE IN FLOOR BESIDE STACK
SIDE ELEVATION OF WARM AIR STACKS "E"
FRONT ELEVATION OF WARM AIR STACKS "F"
FRONT ELEVATION OF COLD AIR STACKS 1-A-B
SIDE ELEVATION OF COLD AIR STACKS 1-A-B

SHEET NO. 2

drain to the furnace room so that at no time could water accumulate and have any effect on the system.

While no analysis has been made, as far as the writer knows, of the humidity in the air upstairs, the theory is advanced that the return supply would pick up a certain percentage of moisture from the walls of the return underground duct and have at least some small influence on the percentage of



SHEET NO. 3

SIZE OPENING AUDITORIUM FLOOR FOR W. A. & C. A. DUCTS

	Warm Air		Cold Air
A	18 in. x 21 in. finished opening in floor	A	12 in. x 30 in. finished opening in floor
B	18 in. x 21 in. finished opening in floor	B	10 in. x 30 in. finished opening in floor
C	14 in. x 26 in. finished opening in floor	C	10 in. x 12 in. finished opening in floor
D	14 in. x 26 in. finished opening in floor	D	18 in. x 22 in. finished opening in floor
E	14 in. x 26 in. finished opening in floor	E	22 in. x 28 in. finished opening in floor
F	10 in. x 18 in. finished opening in floor		

humidity delivered into the church. Where the vertical ducts connect with the underground ducts the joints were made air-tight and the entire system from the return inlet to the warm air outlet was sealed so as to prevent any possibility of dust or dirt getting into the air passages.

Sufficient Capacity Important

While it is oftentimes true that a church board or even an architect will advise the heating contractor that only one part of the building is to be heated at a time, experience proves that this is seldom if ever carried out and it is good business to figure capacity enough to take care of the entire building at all times. Failure of most heating systems in churches can be traced directly to the lack of capacity of the heating system and the necessity for overfiring beyond the safe and ordinary use of the apparatus installed.

The detailed drawings show the many complications that had to be met in the furnace room to bring the duct work to the furnaces without reducing areas or developing too much friction.

Consideration also had to be given to the possible noise developed by the fans and also any noise developed by the air movements. No criticism has been raised on this point since the installation was made.

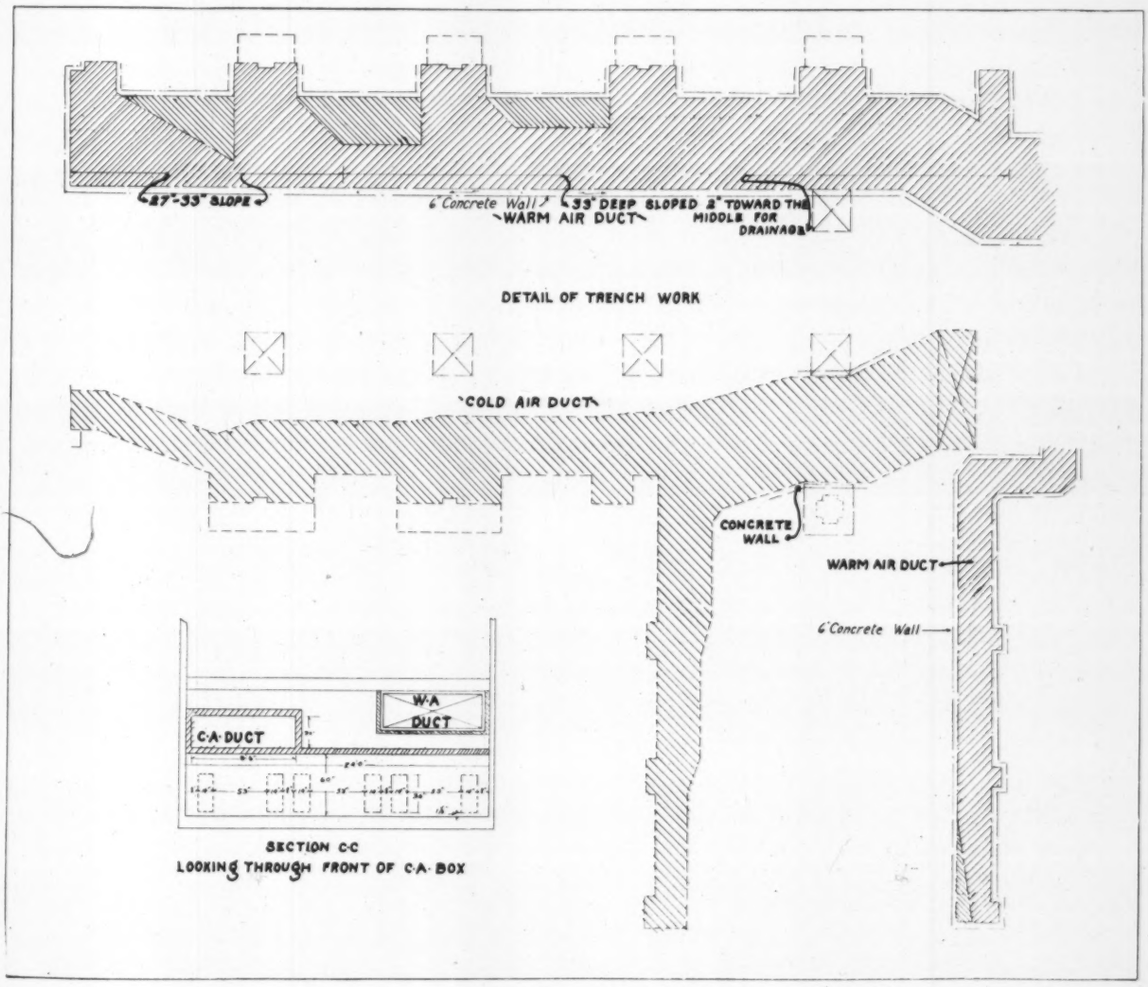
The Heating Plant

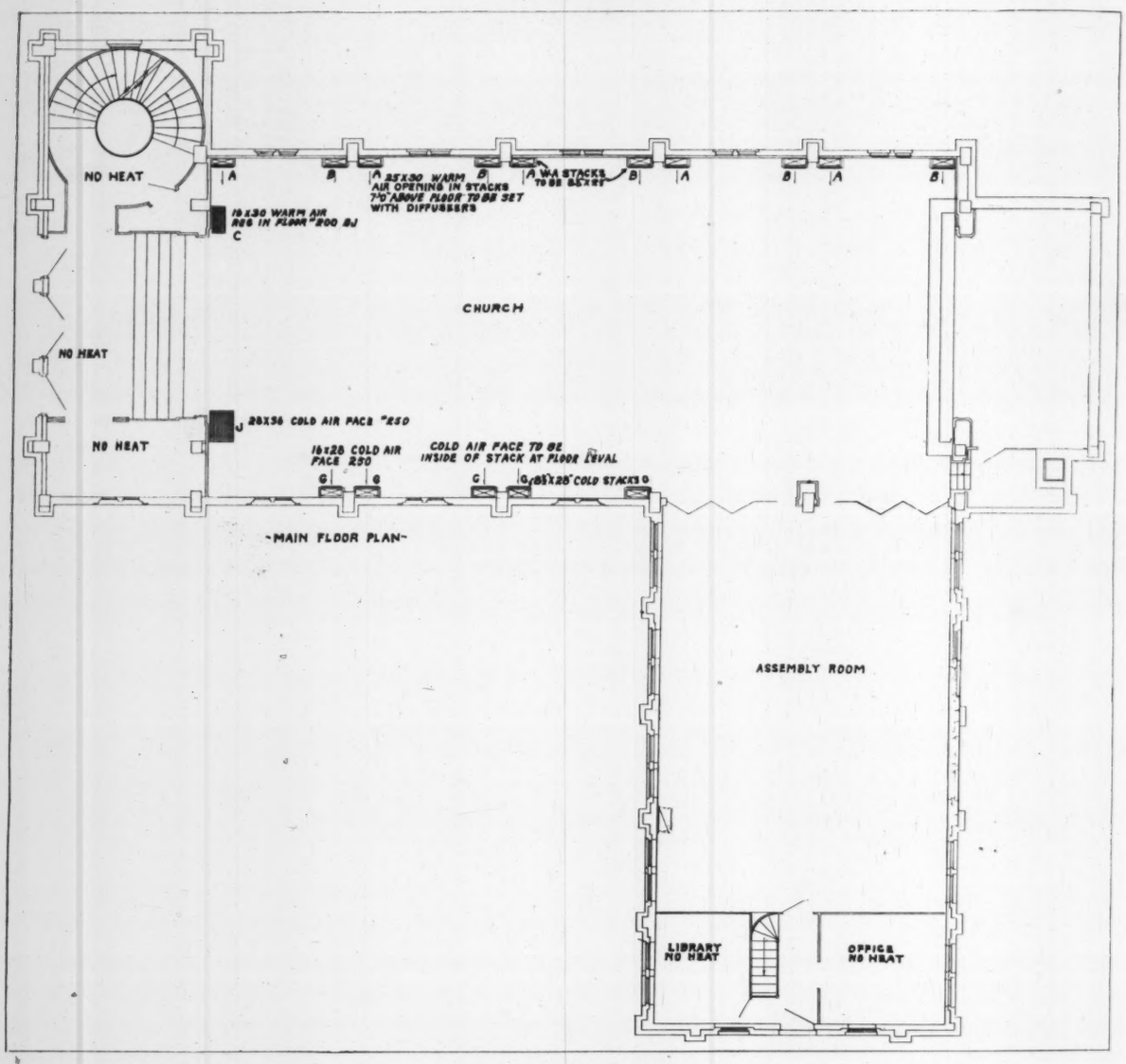
The heating plant consists of three vertical tubular steel furnaces, each equipped with a specially constructed fan placed at the rear of the furnace. The casings around the furnace were carried up on each furnace independent of the other, but the canopy was made to cover all furnaces as a single unit.

The fans consisted each of two 24-in. blades closed in the center and spaced 11 inches apart. Air was drawn from both sides of the fan. Each fan was operated by a ¾ hp. 3 phase 60 cycle motor, each capable of approximately 4,000 cubic feet per minute. They were also supplied with extra pulley and belt so that the fans could be speeded up during the hot weather and the system used for creating a considerable air movement in the building. No attempt was made to cool beyond a possible 1½ degrees, this much of an effect to be secured by the air passing through all cold underground ducts. A flexible canvas connection was made between the fan and each of the casings.

The smoke pipe from each furnace was connected to a trunk and carried through a plenum chamber to a 20 × 20 flue. The effect of carrying this stack through the plenum chamber was to temper the cold air, reduce the flue loss and economize in the use of fuel.

Volume dampers were placed in the canopy so





SHEET NO. 5

SIZES OF OPENINGS IN CHURCH FLOOR FOR C. A. AND W. A. DUCTS

A	10½ in. x 27 in. finished opening in floor	G	18 in. x 30 in. finished opening in floor
B	10½ in. x 27 in. finished opening in floor	I	18 in. x 32 in. finished opening in floor
C	16 in. x 30 in. finished opening in floor	J	28 in. x 36 in. finished opening in floor
D	16 in. x 30 in. finished opening in floor		
E	16 in. x 30 in. finished opening in floor		
F	8 in. x 16 in. finished opening in floor		

that any combination of units could be fired separately or together to heat any definite portion of the church. These dampers also controlled the volume of air sent to the various parts of the building and while under manual control were to be regulated according to the direction of the wind and its velocity.

The outlets in the basement and on the first floor were seven feet in the clear above the floor line. Those on the first floor were equipped with deflecting vanes or diffusers so that regardless of velocities there would not be a blast developed that would create air currents objectionable to the occupants. All risers were boxed in and lathed and plastered and on the first floor were flush with the columns so that no piping was visible.

Computations

The computations consisted of a careful figuring of the glass, the net exposed wall, the roof and the

cubic contents. Three air changes per hour under control were computed and a temperature rise of 50 degrees. The lowest recorded temperature where this job happened to be installed was 15 degrees above zero and the temperature called for 5 feet above the floor was 65 degrees. On account of the height of the ceiling in the main auditorium it was felt that the capacity of the plant should be independent of any temperature rise created by the occupants.

The loss from infiltration was not taken into consideration because of temperatures prevalent in that locality and the capacity of the plant. The total B.t.u. requirements for the building were 1,369,350.

The fuel used was oil and the range within which the burners were to operate varied from 3½ gallons per hour per furnace to six gallons per hour, the lower rate being for ordinary working conditions and the higher rate to take care of extremes in temperature and excess losses caused from infiltration and wind velocities in excess of 15 miles per hour. The efficiency of the burner and unit was based on 80 per cent as determined by flue gas losses when the apparatus was originally under test.

Instructions Important

Safety factors in the computations run as high as 25 per cent and due consideration was given for balancing the system when working under any particular combination.

A complete set of blue prints and working drawings were provided the church for filing for record and complete instructions were issued on how to control, operate and service the entire plant. The latter item is one of the utmost importance, as too often a mechanical job of this character is turned over by the heating contractor without giving proper instructions on its operation and such a system placed in the hands of irresponsible people causes dissatisfaction and trouble to all concerned. When, however, the plant is operated by competent persons under complete instructions, one can be assured that such an installation will last for many years and be a source of satisfaction to the makers, installers and users.

"Open for Discussion"

A department in which we follow the custom of technical societies of allotting space in their programs for discussion of the papers presented

INSULATION AND AIR CONDITIONING

"Refrigeration Limits." *An interview with E. P. Heckel.* Page 169, issue of June, 1929

I HAVE noted the conclusions as to the present possibilities in dealing with the situation in the Madison Square Gardens, as presented in the interview with Mr. Heckel in your June issue.

It would seem that one of the bogies that prevents the solution of such problems is the belief that insulation of such buildings, to bring them up to about cold storage standards, is an economic impossibility.

The fact is, that if the engineer doing a job of this type would impress upon the architect the ridiculousness of the proposition of trying to do the impossible, the architect would in turn exert a more powerful influence on the manufacturers of building materials to produce something that would meet the requirements.

As things are now, however, the architect designs his building, and the engineer proceeds to carry the loads on steel, and building by-laws demand that the panels they fill in between the exterior columns and girders shall be about 12 inches thick, and of materials that have practically no value in resisting the transmission of heat. The question then is, what to add to the inside of such walls to help the engineer design a plant that would make this arena satisfactory under the average condition of occupancy. Such added insulation is, of course, always an extra over the cost of the materials that go to make up the original wall thickness.

The immediate future development will, I think, be the substitution of an insulated backing to a thin veneer of brick, stone, terra cotta, concrete, or whatever you will; this backing to be of such thickness, and to have such insulating value that the combined wall will have a total resistance equal, at least, to the walls used in good cold storage practice.

There are several materials on the market now that could give such a result. For instance, a block, 10 in. thick having a resistance of 1.35, would give a total of 13.5 hours. Add to this the resistance of 4 in. of brick facing, plus interior plaster finish, plus a waterproofing between the brick veneer and the backing block, plus interior and exterior surfaces, and the total would be at least 15 hours. That is to say, it would take one heat unit 15 hours to flow through a square foot under one degree of heat head.

The wall would be about equal to the average cold storage wall having 4 in. of cork. The insulated backing would only be slightly higher in price than the ordinary hollow terra cotta or brick backing which is now used in steel structures, and any slight difference in cost per square foot would be more than offset by the reduction in the size of the refrigerating plant required.

That such a building would go a long way towards meeting the requirements has been proved this year in

Canada where indoor and outdoor temperature records have been maintained in the packing shed of the Sheridan Nurseries Limited at Clarkson, Ontario. There, in spite of considerable areas of glass in walls and roof, and large trucking doors at each end of the building which are open for most of the day, the inside temperature of between 60 and 70 deg. has been maintained for weeks at a time, when the outside recording thermometer showed that the temperature was rising as high as 100 deg. These results have been obtained without any cooling or ventilating plant whatsoever.

The resistance to heat flow built into the walls and roof of this building was about 13 hours or slightly less than the construction outlined for the walls described in this letter. The walls of this building, of course, were not of a type that could be used in the fireproof sections of a large city, but mineral blocks have been made, and can now be made, of a nature to meet fireproof construction requirements, and that would have the necessary resistance to heat flow outlined above.

The whole trouble in this field of work is, that insulation manufacturers have been concerned so much about minute differences of ¼ hour and ½ hour in the total results obtained from their materials, that engineers and architects have had their vision of the real possibilities in construction clouded, and it is only by co-operation between engineers who are attempting to do some real work in air conditioning and heating and the research departments of the larger corporations in the building business, that influence will be exerted on architects to build walls and roofs that will make air conditioning economically feasible for almost any type of structure.

AN ARCHITECT.

An Explanation by E. P. Heckel

The comments in the above letter are well founded. While it is natural that many architects are interested primarily in the architectural design and give insufficient consideration to the mechanical equipment that must be installed to make the building habitable, the architects of the Madison Square Gardens and also the architects of the Chicago Stadium do not belong in that category. Every consideration was paid to the details relating to the mechanical equipment and it is believed that ere long, more architects will realize that it is the duty of both the architect and the engineer to co-operate to the fullest extent in order to achieve maximum results.

The article which was published in Heating, Piping and Air Conditioning was incomplete; it did not tell the entire story. What there was, however, was quite true to the facts. It is not alone the question of insulation that would help to make air conditioning economically feasible for almost any type of structure; there are other factors that must be considered.

For example, we have various city codes that prescribe certain limitations in the matter of ventilation and quantity. In Chicago, they are very strict. The code says that not less than 25 cu. ft. of air per minute from outdoors must be supplied per occupant. Taking this into account, and considering the necessity of cooling, say, 500,000 cu. ft. of air per minute from an outside maximum condition assumed at 95 deg. and 78 deg. wet bulb temperature, to a dewpoint temperature of 43 deg. it can be appreciated what this means in refrigeration tonnage and capacity requirements.

The Chicago Stadium will have 1,000 tons of refrigeration capacity based upon air conditioning load. This is sufficient to cool the stadium, which has a 20,000 seating capacity, to a temperature of around 80 deg. and provide relative humidities of around 55 per cent. To accomplish this stadium cooling and air conditioning requires that the air be first cooled to a dewpoint temperature of 60 deg. before it is delivered to the building. The principal heat load to be taken care of by the cooling air is produced by people and lights, while in proportion only a small amount is due to transmission through walls and roof.

Now if it were desired to cool this building to a temperature of 60 deg. to facilitate ice making on the skating rink, it would mean that this entire fan capacity of air would have to be cooled to approximately 43 deg. and the tonnage requirements would then be nearly 2,000 tons.

With proper insulation on the walls, and with equipment designed to recirculate 100 per cent of the air used for ventilation, a very much reduced refrigeration tonnage would be required to hold the desired temperatures and humidities within the stadium.

Obviously, it can be seen that to use refrigeration for air cooling to provide 60 deg. temperature in summer time to facilitate skating we would require separate refrigeration machines for brine cooling for skating rink purposes and separate refrigeration for air conditioning. This would increase the number of machines now used to almost twice the number installed. At the present time the refrigeration machines installed are used alternately, for air conditioning in summer and for brine cooling in winter.

It is quite certain that few people could be induced to spend several hours in 60-deg. temperature when outside conditions reached summer temperature levels. The experience at Madison Square Garden was cited to illustrate the point discussed in the interview with the Heating, Piping and Air Conditioning editor and not to suggest ice skating in warm weather as a feasible proposition.

The writer would like to ask for a little more information with reference to the conditions mentioned as existing in the Sheridan Nurseries, Limited, at Clarkson, Ontario. It was stated that without any refrigeration whatsoever, the temperatures were noted between 60 and 70 deg. in the packing shed, whereas the outside recording thermometer showed the temperatures rising as high as 100 deg. This cooling effect must have been obtained from some source. It may have been from stored up energy in the form of materials or mass pre-cooled in some other department.

May we illustrate what we have in mind from observations taken in some of the beet sugar warehouses? Some of these warehouses are of capacity to store upward of twelve million pounds of sugar. This sugar is stored in bags, and the warehouses are unheated throughout the entire winter. We have seen these warehouses in June and July, while still half loaded, that registered temperatures around 55 to 60 deg. even though the outside temperatures registered 80 to 85 deg. This interior cold condition can be explained as follows: since the ware-

houses are unheated, the sugar naturally becomes very cold throughout the winter months and stores up a tremendous amount of cold, which is quite frequently as low as 32 or 33 deg. Owing to the lag in cold liberation from this large mass, you can compare the effect with an ice storage plant. This accounts for the cold interior conditions that exist in such warehouses even in July and August.

It seems quite likely that the conditions mentioned in the above letter as existing in the Sheridan Nurseries, Limited, may be due to some such condition as have been observed in beet sugar warehouses.

E. P. Heckel.

SUNSHINE AND DUST

"Outdoor Dust," *by H. C. Murphy.* Page 212, issue of July, 1929

THE article on "Outdoor Dust," by H. C. Murphy, has two tables, the first of which shows the average decrease in sunshine for various cities.

A study of the average sunshine reports of the U. S. Weather Bureau for Cincinnati shows the following percentages:

1915, 59 per cent; 1916, 64 per cent; 1917, 62 per cent; 1918, 57 per cent; 1919, 59 per cent; 1920, 54 per cent; 1921, 56 per cent; 1922, 59 per cent; 1923, 55 per cent; 1924, 56 per cent; 1925, 56 per cent; 1926, 48 per cent; 1927, 60 per cent; 1928, 59 per cent.

Mr. Murphy has selected three months of the year 1926, which established our lowest record, in comparison with the same months of 1916, which showed the highest average. A twenty year average from 1891 to 1910 is 57.1 per cent. An eighteen year average from 1911 to 1928 is 57.2 per cent. This shows a slight average increase against Mr. Murphy's 17 per cent average decrease.

The U. S. Weather Bureau method for determining the percentage of sunshine hours is not a criterion for dust content of the atmosphere. This method merely records the hours of sufficient sunshine to cast a shadow which is practically not sensitive to ordinary atmospheric dust or smoke. What should be shown in connection with atmospheric dust are erythema hours or hours in which the sun's rays will produce a reddening of untanned Caucasian skin. These are the beneficial ultraviolet rays which are obstructed by dust and smoke.

The next table is Mr. Murphy's report on the average dust conditions in the same cities. This table was published in 1927 and is misleading in regard to the city of Cincinnati.

The U. S. Weather Bureau's daily observations on the number of dust particles per cubic foot of Cincinnati atmosphere shows a wide variation on different days because of weather, wind and other conditions. The operation of making the dust count is extremely technical. A few observations in some particular locality are by no means average conditions.

C. M. Stegner,
Director of Buildings, Cincinnati, O.

Mr. Murphy Replies

Taking the points in Mr. Stegner's letter in the order in which they are made, it may be that he is entirely correct in his remarks as to the figures obtained and methods used by the United States Weather Bureau for measuring sunshine. The fact remains, however, that the Marvin Sunshine Recorder is the only practical method of recording sunshine we have and its results are generally accepted throughout the world.

While the instrument used in my work is admittedly not perfect, it is quite generally accepted as the most practical instrument for field work now available. Instruments of this design, that is, the jet impingement type of dust counter, have been in use in practically every civilized country in the world since 1922. Under the provision made by the International Union for Geodesy and Geophysics in Rome in 1922, jet dust counters were sent to the following official bodies and results are tabulated by various public agencies throughout the world:

Meteorological Office	Rome
Meteorological Office	Paris
Meteorological Office	Stockholm
Meteorological Office	Japan
Meteorological Office	Lisbon
Meteorological Office	Bucharest
Meteorological Office	Poland
Meteorological Office	Athens
Meteorological Office	London
Meteorological Office	Toronto
Meteorological Office	Rio de Janeiro
Meteorological Office	Madrid
Weather Bureau	Washington
Baron de Dorlodot	Belgium
Meteorological Bureau	Melbourne, Australia
Geophysical Observatory	Pavia, Italy

More recently the United States Weather Bureau has equipped its offices in Cincinnati and twelve other cities with instruments of this type and daily records of the dust conditions are being made.

As stated in Mr. Stegner's letter, the operation of making the dust count is extremely technical and until the observers have had considerable practice there will be a wide variation in the results. It has been amply demonstrated, however, that with a standardized procedure and continuous practice, entirely consistent results can be had. The necessity of a standardized technique is well appreciated and at the suggestion of the United States Weather Bureau officials, the writer and others skilled in the use of these instruments personally visited the weather bureaus in different cities and instructed them in the methods of making the tests.

As regards the average percentages of sunshine quoted in his letter from the weather bureau at Cincinnati, these are yearly averages. My figures are the published figures of the U. S. Weather Bureau and, as stated in my article, are for the three winter months of December, January and February—in the height of the heating season, the months when preventable smoke and dust do the most harm. Yearly reports of the U. S. Weather Bureau show that the year 1926 was not the lowest average sunshine record for the winter months. Even if this were the case, however, it could not alter the relative position of Cincinnati among the ten cities noted in my survey, as the years 1916 and 1926 were used as a

basis in every case. I selected the year 1926 for my intensive study and the same period ten years previously because at the time of my investigation the year 1926 was the latest year for which the published data for all the cities studied were available.

H. C. MURPHY.

Piping Locomotive Washing Plant

When a locomotive arrives at a division terminal after a long run, it is standard practice on American railways to blow out the boiler and wash it out with hot water. A schematic layout of one of the boiler washing plants of the Santa Fe railroad is shown in the accompanying illustration. The drawing shows three locomotives in the following order:

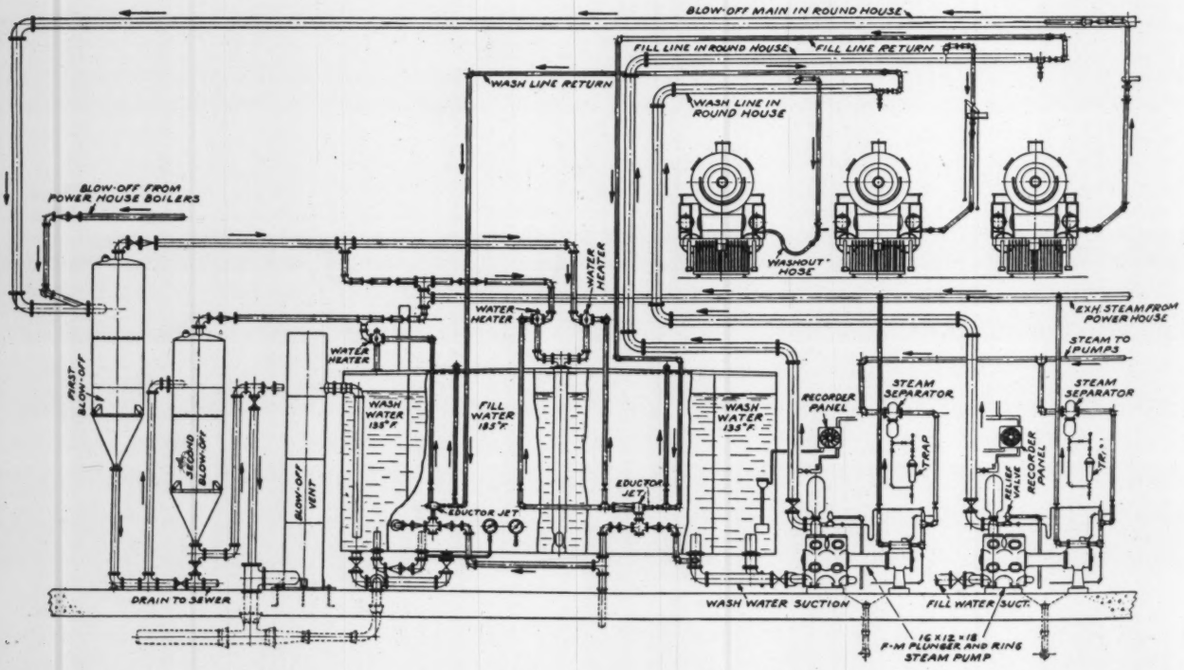
The engine on the right hand side has just arrived at the roundhouse from its scheduled run. The hot water and steam remaining in the boiler of the locomotive are blown off through one of three drops which are hung in each stall, the three lines being known as No. 1, the blow-off line; No. 2, the washout line; and No. 3, the refill line.

In detail, the operation of the system is as follows,—the blow-off line from the roundhouse and the blow-off from the power house empty into the first blow-off tank, the hot water passing out of the bottom of the tank through a 6-in. pipe to the second blow-off tank. The steam from the first blow-off tank passes out the top and through a check valve to a water heater. At this point the steam is condensed as it unites with water delivered by an eductor jet. The water comes to the jet either from the make-up supply or from the bottom of the fill tank. The heater is also supplied with steam from the power house exhaust which flows to the heater through a check valve when no steam is coming from the first blow-off tank. The jet is actuated by water at 150 lb. pressure from the hot water fill return line which is taken from the end of the hot water fill in the roundhouse.

The second heater is supplied with steam from the power house exhaust and with water from the jet just described. The water discharges from the heaters into the large pipe shown in the center of the fill water tank. Blow-off water from the first blow-off tank entering the second blow-off tank passes out at the bottom of the tank into the sewer. The heater for the wash water is supplied with steam from the second blow-off tank and also from the power house exhaust. This steam is condensed as it unites with the water delivered to the heater by the eductor jet. This water comes to the jet either from the make-up supply or from the bottom of the wash tank. This jet is also actuated by water at 150 lb. pressure from the hot water wash return line which is taken from the end of the wash line in the roundhouse. The water discharges from the heater into the wash tank by means of the vertical connection shown between the heater and the top of the wash tank.

It will be noted that the fill water and wash water tanks are concentric. That is, the fill water tank is located inside the wash water tank. The steam from the first blow-off tank, therefore, not only heats the fill water to a temperature of 185 deg. fahr. but the water in this fill tank also helps raise the temperature of the wash water to the required 135 deg. fahr.

The two 16 by 12 by 18-in. steam pumps take their suction from the bottom of the wash water and fill tanks as shown and discharge into the roundhouse mains. The pressure is maintained at a uniform point by the pressure governors. There are also miscellaneous valves used for draining the blow-off tanks, shutting off the suction on the pump, and similar purposes.



INTERVIEWS OF INTEREST

DR. DAVID C. TODD, school board member, St. Louis, who for more than a year has been directing a study of environmental conditions in St. Louis' special schools, describes in a recent interview the manner in which his committee has approached the subject of school ventilation.

"The first part of our inquiry related to air conditions in teachers' colleges, and our first critique called attention to the different standards that prevailed in our white and colored schools," states Dr. Todd. "It was some months later that the committee was empowered to look into the whole matter of the conduct of our special schools.

"Certain health groups had objected that the special rooms maintained for pre-tuberculous children were not in the strictest sense 'fresh air' rooms, and had asked that here at least the air supply of class rooms should be untreated, unfiltered, unconditioned, and, in fact, *un*-everything else that savored of artificial method.

"Naturally, the school board found it impossible to meet the issues involved, or to recommend changes in any given direction without exhaustive inquiry, and it was at this point that the functions of our committee were enlarged to include ventilation studies in St. Louis schools as a whole.

"Another critique for which the committee is responsible was that the Taussig School for pre-tuberculous children depends for heat upon a big pot stove that at times creates great extremes within the room, and always results in unequal conditions that may represent really serious problems in adaptation.

"We then attempted to measure the actual health values of open air schools under conditions that prevail in St. Louis climate. Open air in a wind-swept western plain means vital air, dust-free, smoke-free, and charged with natural ozone. What does open air mean in St. Louis?

"Despite the elaboration of ventilation facilities that has been developed in St. Louis, we made the remarkable observation at this point that no one knew just what are the physical characteristics of the air finally delivered from our conditioning apparatus. Air washers are routinely installed that are bought on a basis of 95 to 99 per cent efficiency for the removal of solid impurities. 'We know that the washers are efficient,' declared the engineers; 'just look at the filth the washers remove from the air.' "

"Of course, it is not dirt removed that is important, but the percentages of dust that remain. And a third surprise awaited us here. In certain schools the actual performance of air washers capable of 98 per cent dust removal ranged around 37.7 per cent. We went to an engineer for advice. He instituted a thorough-going cleaning up of air washers and other apparatus, after which the efficiency was 93.2 per cent in the washers instead of the 37.7 per cent previously observed. The committee's critique at this point has resulted in new operating schedules for air washers, and consistent operating supervision.

"St. Louis type of air does not favor open windows for schools. It is a manufacturing center. The topography is low. Barometric conditions bring gas and dust down to the surface and keep them there. We get all of the impurity there is. There is nearly always a demonstrable amount of sulphur dioxide. We have ruled, therefore, that in St. Louis at least, the schools for health promotion shall have conditioned air. Our special problem demands it.

"The test reports demonstrate the effectiveness of washing St. Louis' air:

Scullin School

Before Washing	After Washing
Dust count 470-200,500 particles per cubic foot.	32-13,650 dust particles per cubic foot.
Efficiency of dust removal, 93.2 per cent.	
Sulphur dioxide—Trace.	No trace.

Lincoln School

Before Washing	After Washing
Dust count 340-145,000 particles per cubic foot.	43-18,350 dust particles per cubic foot.
Efficiency of dust removal, 87.5 per cent.	
Sulphur dioxide—Trace.	No trace.

"Our attempt to secure a further check by means of bacteriological counts was soon abandoned. Re-contamination from the presence of so many individuals occurred to such an extent that no reliable data could be gathered. Nevertheless, on the basis of physical evidence alone from tests of air at inlet and outlet of air washers the proof was conclusive that the new set up for air hygiene in St. Louis schools must include conditioned air.

"The committee then went a-traveling. We visited Toronto, Cleveland, Detroit. We studied vocational schools. We scrutinized ventilation practices. We talked to experts on humidity ranges, mean temperatures, the effect of sunlight, and artificial illumination.

"The final word of the committee has not yet been enunciated, but St. Louis may be said to have worked out her own definitions of air hygiene, and decision has been reached that hereafter the care that has been provided for pre-tuberculous children alone will be developed still further into a broadly inclusive health promotion work with full responsibility assumed by the board of education for its proper conduct. These rooms will have conditioned air. The school for cripples will become a vocational school of the most approved type where handicaps are corrected or compensated for according to the most highly scientific and humanitarian methods possible.

It is a broad program, broadly conceived, and the air environment is not the least important factor in its development."

Measurement of Flow of Liquids by the Co-ordinate Method

By Prof. F. W. Greve

THE CO-ORDINATE method of measuring pipe discharge depends upon the principle that a definite relation exists between the rate of flow within a horizontal pipe and the co-ordinates of the stream issuing therefrom. This method presents certain elements of convenience, such as simple and inexpensive apparatus, is quickly and cheaply operated, and is especially advantageous in connection with flows containing sediment and other suspended matter. Where solids are moving along the bottom of a pipe at a speed less than that of the liquid, it becomes necessary to determine the ratio of the two velocities.

The co-ordinate method has been used where the approximate discharge is required and also in certain other cases where instruments of precision such as venturi meters, pitometers, water meters, and other meters are impracticable because of the clogging action of suspended matter in the water. It has been applied to the free discharge from sewers, pipes, pumps, wells, dredges, sluices, etc., and provides the basis for the design of leaping weirs. However, it is obvious from a review of the literature on the subject that there are few reliable data pertaining to necessary empirical constants.

The co-ordinate method of determining pipe flow is simple, inexpensive and capable of many applications in field tests where a maximum error of 10 per cent is permissible. The author, who conducted the tests described in this article, is professor of hydraulic engineering, Purdue University, Lafayette, Ind.

The investigation was conducted in the hydraulic laboratory of Purdue University to determine (1) the empirical relation between the rate of discharge from a horizontal pipe and the co-ordinates of the upper surface of the issuing stream or jet, measured from the top, inside of the pipe at the point of discharge; (2) the relation between the actual mean velocity within the pipe to that indicated by the law of a freely falling body in vacuo as applied to the upper surface of the jet; and (3) the possibilities and limitations of the method, especially as to the effect of the personal element in taking measurements.

Grateful acknowledgment is made to Dr. M. J. Zucrow and M. P. O'Brien, members of the Engineering Experiment Station staff, for their able assistance both in the performance of the tests and in the computations of data.

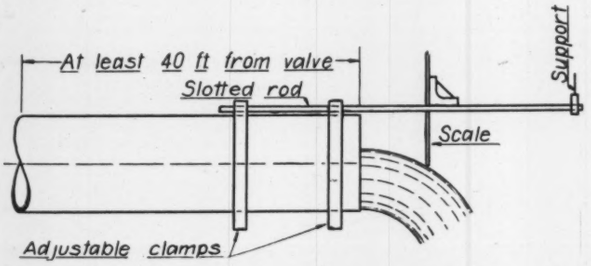


FIG. 1--LAYOUT OF THE APPARATUS USED IN MEASURING PIPE DISCHARGE BY THE CO-ORDINATE METHOD AT PURDUE

The Apparatus Used

The nominal and true diameters of the pipes used in the investigation are noted in Table 1. The minimum length of horizontal pipe under test was 40 feet, the flow thereto being supplied by centrifugal pumps and controlled by valves. Movable chutes served to direct the discharge into calibrated weighing tanks during a test and to shunt it back into the reservoir between tests. Calibrated stopwatches were used to observe time-intervals.

A machinists' combination square and a flat steel rod fastened to the top of a pipe in a position parallel to its axis were used to measure the co-ordinates of a selected point in the upper surface of the issuing stream. The protruding portion of the rod, divided into units of one-half foot, was supported at the unattached end by a metal sling and turnbuckle whereby the rod could be maintained precisely level. The steel rod was replaced, after the trial operation leg one of machined steel, slotted, for the insertion of the scale and fixed to the pipes by adjustable steel clamps. The layout of the apparatus as finally used is diagrammatically illustrated in Figure 1.

Eight readings of the true diameter of each pipe were obtained by calipers, the average value being as listed in Table 1. A given pipe was then suspended from the ceiling of the main floor and aligned horizontally with either a spirit or a precise level. The rod was clamped in position, parallel to the axis of the pipe, and adjusted horizontally with the turnbuckle at its projecting end. Great care was taken in the setup to insure placement of the scale for measuring the ordinate of the jet in a vertical plane through the axis of the pipe.

The upper surface curve of the jet was outlined

by scale readings from the top of the rod, at one-half foot horizontal intervals from the end of the pipe. The former measurements are designated u, and the latter x.

The recorded data per test consisted of from five to ten readings of u for each value of x. Values of u were read to sixty-fourths of an inch. The discharge and x remained constant for each test, the time ranging from two to twenty minutes depending upon the amount of flow. The rate of discharge was varied from a minimum so that the ratio of depth to diameter at the outlet, K, was equal to about twenty per cent, to a maximum which occurred when u was approximately one foot with x equal to two feet. The volume of efflux from the two-inch pipe was weighed in a tank of 4,000 pounds capacity, while that from the other pipes was similarly measured in a tank of 20 tons capacity. Both tanks were calibrated previous to the experiments.

The mean weight of a cubic foot of water was

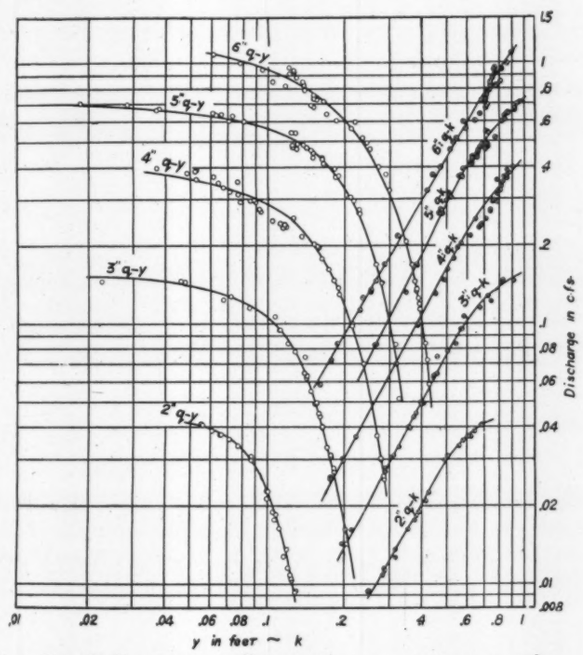


FIG. 2—RELATION OF DISCHARGE TO Y AND K; X = 0

found to be 62.32 pounds for a temperature range during the experiment of 60 to 78 degrees fahrenheit.

The surface of a stream issuing from a pipe is not sharply defined, with the result that the correct value of u rests to some extent upon individual judgment. It was for the purpose of determining the effect of this personal element that the investigation was first conducted with the aid of Dr. M. J. Zucrow and later repeated with the assistance of M. P. O'Brien.

Application of the Method

The co-ordinate method of measuring pipe flow originally came into use to enable engineers to solve problems in which approximate rates of flow were desired at a minimum of financial outlay or in which the velocity could not be measured with instruments ordinarily used for that purpose. It is essential for the application of this method that the end section of the pipe

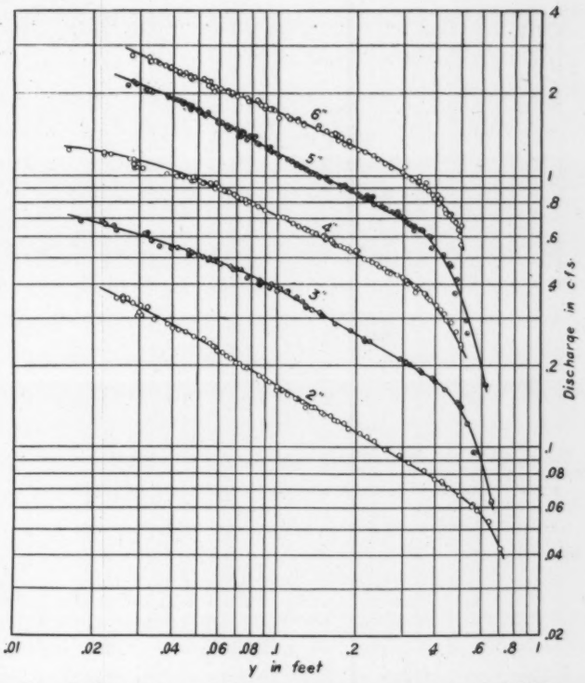


FIG. 3—RELATION OF DISCHARGE TO Y AND K; X = 0.5 FT.

be horizontal and that means be available to measure the co-ordinates of the issuing stream, such as a straight edge and scale or a square.

For instance, a series of experiments were conducted to determine the velocity of discharge from dredge pipes in which silt, sand, and gravel comprised a considerable portion of the efflux. In another application there were recorded the results of certain experiments made to determine the discharge from one of eight scouring sluices located in the Krishuarajasagara dam that spans the

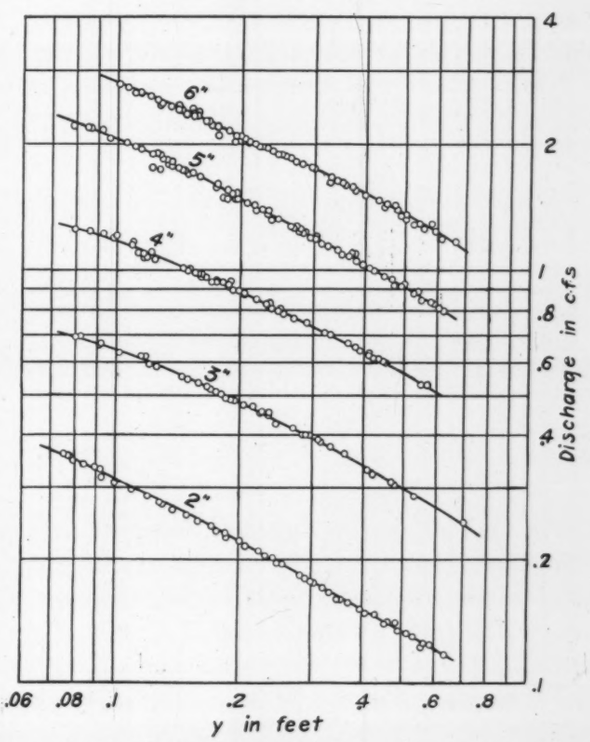


FIG. 4—RELATION OF DISCHARGE TO Y; X = 1 FT.

Cauvery river in southern India. The sluice was inaccessible, thus prohibiting actual measurement of the co-ordinates of the discharging stream or the use of current meters and gages. Calculations or discharge were based upon the values of the co-ordinates as scaled from an enlarged photograph taken from one end of the dam.

In the tests conducted at Purdue, the co-ordinates were measured to the upper surface of the issuing stream with the origin of the axes situated at the point where this upper surface curve emerged from the pipe or sluice. A particle of water falling through the atmosphere will follow the path of a parabola and its theoretical velocity at any point can be computed as previously outlined. To this theoretical velocity, there must be applied a coefficient in order to correlate its value with the actual mean velocity within the pipe. The coefficients can only be determined empirically.

In an article entitled, "The California Method of Pipe Flow," is described a series of tests made in the laboratory of the University of California.

The writer has made use of the co-ordinate method on two occasions. In the first instance, it was necessary to determine the efficiency of a dredge pump and the friction loss in the 6-inch piping system for various discharges. Pressures were read on Bourdon gages. Rates of flow for the measured co-ordinates at the discharge end of the pipe were interpolated from curves similar to those for the 6-in. pipe shown in Figs. 3, 4, 5, and 6. In the second instance, the method was applied to measure the flow from a well that formed part of the water supply for a small municipality. As illustrated in Fig. 7, the flow was insufficient to fill the pipe. The ratio of depth to diameter was calculated at the pipe end and the discharge was noted on a curve similar to those shown in Fig. 2.

Table 1. Sizes of Pipe

Nominal diameter, inches..	6	5	4	3	2
True diameter, feet.....	.504	.423	.342	.255	.1714

TABLE 2—DISCHARGE, Q, IN C.F.S. WHEN x = 0

K = Depth / Diameter	6-in. pipe		5-in. pipe		4-in. pipe		3-in. pipe		2-in. pipe	
	Depth ft.	Discharge c.f.s.	Depth ft.	Discharge c.f.s.	Depth ft.	Discharge c.f.s.	Depth ft.	Discharge c.f.s.	Depth ft.	Discharge c.f.s.
.15			.0635		.0514		.0382		.0260	
.20	.101	.0942	.0847		.0685	.0309	.0509	.0138	.0347	
.25	.126	.136	.106	.074	.0857	.0461	.0637	.0210	.0434	.0088
.30	.151	.185	.127	.106	.103	.0644	.0764	.0294	.0521	.0122
.35	.176	.240	.148	.145	.120	.0856	.0891	.0393	.0607	.0160
.40	.202	.301	.169	.190	.137	.108	.102	.0504	.0694	.0203
.45	.227	.368	.190	.240	.154	.135	.115	.0630	.0781	.0250
.50	.252	.440	.212	.297	.171	.163	.127	.0772	.0868	.0301
.55	.277	.517	.233	.354	.188	.193	.140	.0920	.0954	.0341
.60	.303	.598	.254	.410	.206	.223	.153	.103	.104	.0370
.65	.328	.683	.275	.461	.223	.253	.166	.114	.113	.0392
.70	.353	.770	.296	.508	.240	.282	.178	.123	.122	.0410
.75	.378	.856	.317	.550	.257	.308	.191	.132	.130	
.80	.403	.943	.339	.588	.274	.334	.204	.138	.139	
.85	.429	1.02	.360	.622	.291	.358	.217	.143	.148	
.90	.454	1.10	.381	.652	.308	.380	.229	.148	.156	
.95	.479		.402	.680	.325	.400	.242	.152	.165	

TABLE 3—DISCHARGE, Q, IN C.F.S. AND C WHEN x = 0.5 FT.

y ft.	Nominal Diameter in Inches									
	2		3		4		5		6	
	q	C	q	C	q	C	q	C	q	C
.02	.395	1.20	.69	.94	1.22	.93				
.03	.326	1.22	.61	1.02	1.12	1.05	2.16	1.32	2.77	1.20
.04	.280	1.21	.55	1.08	1.03	1.12	1.91	1.35	2.48	1.24
.05	.247	1.19	.51	1.12	.97	1.17	1.72	1.36	2.27	1.27
.06	.223	1.18	.48	1.15	.90	1.19	1.57	1.36	2.11	1.29
.07	.204	1.17	.45	1.17	.84	1.20	1.44	1.35	1.98	1.31
.08	.190	1.16	.43	1.19	.79	1.21	1.35	1.36	1.88	1.33
.09	.177	1.15	.41	1.19	.75	1.22	1.28	1.36	1.80	1.35
.10	.167	1.14	.39	1.20	.71	1.22	1.21	1.36	1.72	1.36
.15	.133	1.14	.31	1.17	.59	1.24	1.00	1.36	1.47	1.42
.20	.113	1.09	.265	1.16	.51	1.24	.87	1.37	1.30	1.45
.25	.100	1.08	.234	1.14	.46	1.25	.78	1.37	1.17	1.46
.30	.090	1.06	.210	1.12	.42	1.25	.70	1.37	1.06	1.45
.35	.083	1.05	.192	1.11	.377	1.21	.62	1.30	.96	1.41
.40	.077	1.05	.175	1.08	.337	1.16	.53	1.19	.86	1.35
.45	.071	1.03	.158	1.03	.297	1.08	.43	1.02	.74	1.24
.50	.066	1.01	.140	.97	.258	.99	.335	.84	.55	.97
.55	.061	.98	.112	.81	.220	.89	.25	.66		
.60	.056	.94	.085	.64	.185	.78				
.65	.051	.89	.064	.50	.153	.67				
.70	.045	.81								

TABLE 4—DISCHARGE, Q, IN C.F.S. AND C WHEN x = 1 FT.

y ft.	Nominal Diameter in Inches									
	2		3		4		5		6	
	q	C	q	C	q	C	q	C	q	C
.08	.35	1.07	.71	.98	1.27	.98	2.26	1.13		
.09	.33	1.07	.68	1.00	1.22	.99	2.17	1.15	2.93	1.10
.10	.31	1.06	.65	1.00	1.18	1.00	2.06	1.16	2.80	1.11
.15	.253	1.06	.55	1.04	.99	1.04	1.70	1.17	2.35	1.14
.20	.220	1.06	.48	1.05	.88	1.07	1.46	1.16	2.07	1.16
.25	.194	1.05	.43	1.05	.80	1.08	1.30	1.15	1.88	1.18
.30	.177	1.05	.392	1.05	.74	1.10	1.18	1.15	1.72	1.18
.35	.162	1.04	.358	1.04	.68	1.10	1.09	1.14	1.60	1.19
.40	.151	1.03	.333	1.04	.64	1.10	1.02	1.14	1.50	1.18
.45	.141	1.02	.312	1.02	.60	1.09	.95	1.13	1.41	1.18
.50	.133	1.02	.295	1.02	.57	1.09	.90	1.13	1.33	1.18
.55	.127	1.02	.280	1.01	.54	1.09	.86	1.13	1.28	1.18
.60	.120	1.00	.267	1.01	.52	1.09	.82	1.13	1.22	1.18
.65	.115	1.00	.254	1.00	.50	1.09	.79	1.13	1.17	1.18

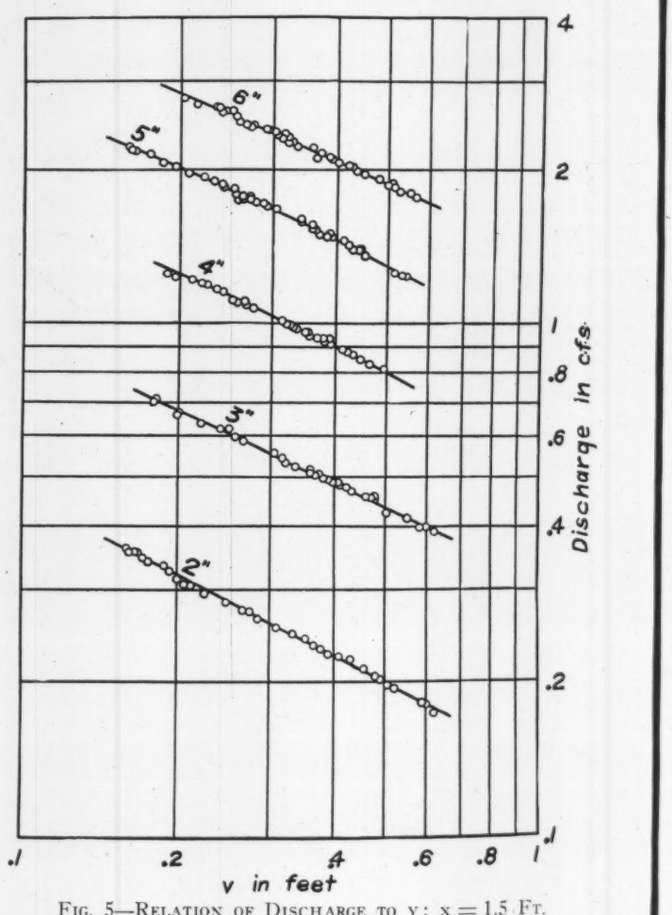


FIG. 5—RELATION OF DISCHARGE TO Y; x = 1.5 FT.

Computations and Results

The dependence of the rate of discharge upon values of x and y is graphically illustrated in Figs. 2 to 6, for

which purpose logarithmic cross-section paper was chosen for the sake of convenience. Fig. 2 also shows the fluctuation in the discharge with variation in the ratio of depth to diameter, K, at the outlet. From graphs similarly plotted to a large scale were interpolated the results set down in Tables 2 to 6.

It will be observed in Fig. 2 that when K is less than 60 per cent, the graphs are straight lines and the pipes are flowing partially full for their entire lengths. No readings were taken when K was less than 20 per cent because under such conditions the jet failed to spring clear of the pipe outlet. For values of K greater than 60 per cent and less than unity, the pipes were completely filled except for a short distance back from the outlet, such distance depending upon K. That is, a sudden transition in the hydraulic radius occurs when the condition of flow changes from a partially to a completely filled pipe, illustrated by the curvature of the graphs for values of K above 60 per cent. That the part-full condition holds true for a pipe only up to an approximate value of 60 per cent for K was verified by supplementary tests using water dyed blue and glass tubes of 65 mm. diameter.

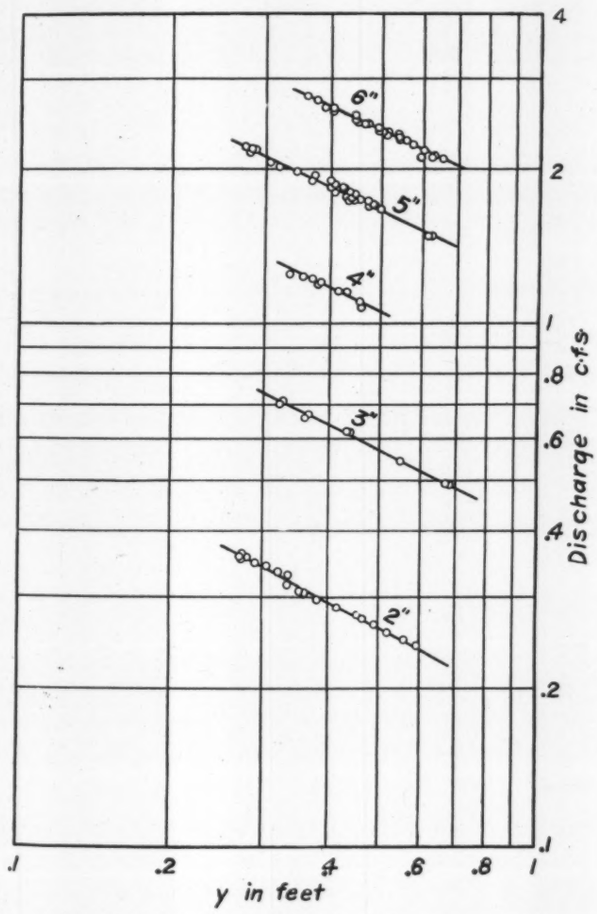


FIG. 6—RELATION OF DISCHARGE TO y; x = 2 FT.

The comparison of the discharge in cubic feet per second, q, with ratio of depth to diameter, K, for partially filled pipes was made in an endeavor to obtain a relation between these two quantities independent of the pipe size. The equations of the q-K graphs in Fig. 2, with 0.20 and 0.60 as the limits of K, can be expressed as

$$q = M(K)^N \quad [1]$$

where M is a coefficient that denotes the value of q when K is unity and N is the slope of the graph with the horizontal plane. The values of M and N were computed by the center of gravity method. Briefly, this method consists of locating the center of gravity of all the plotted points. The points on either side of this center are then

TABLE 5—DISCHARGE, Q, IN C.F.S. AND C WHEN x = 1.5 FT.

y ft.	Nominal Diameter in Inches									
	2		3		4		5		6	
	q	C	q	C	q	C	q	C	q	C
.15	.370	1.03	.77	.97	1.39	.98	2.26	1.04	3.12	1.01
.20	.320	1.03	.68	.99	1.24	1.00	2.02	1.07	2.81	1.05
.25	.287	1.03	.61	.99	1.12	1.01	1.84	1.09	2.57	1.07
.30	.261	1.03	.56	1.00	1.03	1.02	1.68	1.09	2.38	1.09
.35	.242	1.03	.52	1.00	.96	1.03	1.56	1.09	2.21	1.09
.40	.225	1.02	.49	1.00	.90	1.03	1.46	1.09	2.08	1.09
.45	.211	1.02	.46	1.00	.85	1.03	1.37	1.09	1.97	1.09
.50	.199	1.01	.44	1.00	.81	1.03	1.29	1.08	1.87	1.10
.55	.188	1.00	.417	1.01	.77	1.03	1.22	1.07	1.78	1.10
.60	.180	1.00	.400	1.01	.74	1.03	1.16	1.06	1.70	1.10
.65	.172	1.00	.383	1.00	.71	1.03	1.11	1.06	1.63	1.09
.70	.166	1.00	.370	1.00	.68	1.03	1.06	1.05	1.57	1.09

TABLE 6—DISCHARGE, Q, IN C.F.S. AND C WHEN x = 2 FT.

y ft.	Nominal Diameter in Inches									
	2		3		4		5		6	
	q	C	q	C	q	C	q	C	q	C
.25	.375	1.01					2.31	1.03		
.30	.340	1.01	.73	97	1.34	1.00	2.13	1.03	3.02	1.03
.35	.314	1.00	.68	.98	1.27	1.02	1.97	1.03	2.81	1.04
.40	.293	1.00	.63	.98	1.17	1.00	1.85	1.04	2.63	1.04
.45	.275	1.00	.60	.98	1.10	1.00	1.74	1.04	2.50	1.05
.50	.260	.99	.57	.98	1.05	1.01	1.65	1.04	2.37	1.05
.55	.248	.99	.54	.98			1.58	1.04	2.27	1.05
.60	.237	.99	.52	.98			1.52	1.04	2.18	1.05
.65	.227	.99	.50	.98			1.46	1.04	2.09	1.05
.70	.218	.99	.48	.98					2.01	1.05

TABLE 7—VALUES OF M IN $q = M[K]^N$

Nominal diameter, in inches	2	3	4	5	6
M	.110	.272	.591	1.01	1.71

TABLE 8—COMPUTED DISCHARGE IN C.F.S. WHEN x = 0

Nominal Diameter, Inches	K, Depth Diameter	Discharge			
		Actual c.f.s.	Computed c.f.s.	Difference, c.f.s.	Difference, Per Cent
2	.2		.0054		
	.3	.0122	.0114	−.0008	− 6.6
	.4	.0203	.0193	−.001	− 4.9
	.5	.0301	.0291	−.001	− 3.3
	.6	.0370	.0407	+.0037	+10.0
3	.2	.0138	.0149	+.0011	+ 8.0
	.3	.0294	.0315	+.0021	+ 7.1
	.4	.0504	.0535	+.0031	+ 6.1
	.5	.0772	.0806	+.0034	+ 4.4
	.6	.103	.113	+.01	+ 9.7
4	.2	.0309	.0308	−.0001	− 0.3
	.3	.0644	.0648	+.0004	+ 0.6
	.4	.108	.110	+.002	+ 1.8
	.5	.163	.166	+.003	+ 1.8
	.6	.223	.232	+.009	+ 4.0
5	.2		.0538		
	.3	.106	.113	+.007	+ 6.6
	.4	.190	.193	+.003	+ 1.6
	.5	.297	.291	−.006	− 2.0
	.6	.410	.407	−.003	− 0.7
6	.2	.0942	.0851	−.0091	− 9.7
	.3	.185	.179	−.006	− 3.2
	.4	.301	.305	+.004	+ 1.3
	.5	.440	.459	+.019	+ 4.1
	.6	.598	.642	+.044	+ 7.4

treated separately to locate their respective centers of gravity. Inasmuch as a straight line on logarithmic cross-section paper expresses the above law represented by the plotted data, then the three centers of gravity will all lie on the same straight line, the slope of which is necessarily constant and represented by the exponent, N. Since,

$$\log q = \log M + N \log K \quad [2]$$

then

$$N = \frac{\log q_2 - \log q_1}{\log K_2 - \log K_1} = \frac{\log q_3 - \log q_2}{\log K_3 - \log K_2} = \frac{\log q_3 - \log q_1}{\log K_3 - \log K_1} \quad [3]$$

Also,

$$\log M = \log q_1 - N \log K_1 = \log q_2 - N \log K_2 = \log q_3 - N \log K_3 \quad [4]$$

If M and N can be correlated with the diameter, d, then the variation in q with change in K can be formulated without regard to the pipe size. No fixed relation could be found between N and d, but since the change in the former was not large, it was assigned an average value of 1.84. With this constant N, values of M were computed for each pipe, as listed in Table 7, below, and when plotted against values of d on logarithmic crosspaper produced a straight line of the equation.

$$M = R[d]^S \quad [5]$$

The valuation of R and S was made similarly to that for M and N and the equation of the discharge from a partly filled pipe is

$$q = R[d]^S[K]^N \quad [6]$$

Computed rates of discharge are compared with experimental values in Table 8, wherein it is seen that the results obtained by formula are correct to within 10 per cent.

The rule for finding the speed of a freely falling body in vacuo has sometimes been invoked to calculate the theoretical velocity, V, of a particle in the upper surface of the jet, such computed velocity being then multiplied by an empirical coefficient, C, to obtain the actual mean velocity of flow within a pipe, v. Thus,

$$y = \frac{gt^2}{2} \quad [7]$$

$$x = Vt \quad [8]$$

$$v = CV \quad [9]$$

where t is the time in seconds and x and y are co-ordinates of the jet in feet.

The coefficient, C, for values of x greater than zero is evaluated in Tables 3 to 6. No general equation could be found to express its relation to x, y, or the rate of discharge. The degree of variation in C decreases with increase in x, becoming practically nil when x is equal to or greater than one and one-half feet.

FIG. 7—THE CO-ORDINATE METHOD USED TO MEASURE THE FLOW FROM A WELL WHICH FORMED PART OF A MUNICIPAL WATER SUPPLY SYSTEM. NOTE THAT THE FLOW DID NOT FILL THE PIPE

Conclusions

The co-ordinate method of determining pipe flow is simple, inexpensive, and practical for field tests where a maximum error of 10 per cent is permissible.

The discharge in cubic feet per second, q, from partially filled pipes with limiting nominal diameters of two and six inches and with a ratio of depth to diameter at the outlet between 0.2 and 0.6, can be computed from the equation,

$$q = 9.43[d]^{2.56}[K]^{1.84} \quad [10]$$

where d is the true diameter in feet, and

$$q = 0.0181[d]^{2.51}[K]^{1.84} \quad [11]$$

where d is the nominal diameter in inches.

The ordinate, y, increases directly with the abscissa, x, with constant rate of discharge, and therefore the effect of an error in measuring the former distance decreases as the latter is extended. However, at values of x greater than two feet, the stream disintegrates and the data become inaccurate.

Tabulated values of the rate of discharge listed herein will be found more nearly correct and more convenient to handle than computations invoking the application of the law of a body falling freely in vacuo to the jet. The coefficient, C, in the equation for calculating the mean actual velocity of pipe flow in feet per second, $v = C\sqrt{2gy}$ will not vary more than five per cent from unity when x equals two feet.

Heating·Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

SEPTEMBER 1929

Material in this Section prepared by Publication Committee: A. C. Willard, *Chairman*; N. W. Downes, and W. A. Rowe. : A. V. Hutchinson, Manager of Publications

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Preventing Condensation on Interior Building Surfaces

By Paul D. Close[1], New York
MEMBER

OF the many problems with which the engineer is confronted, one of the most important is that of preventing condensation on ceilings and other interior surfaces of buildings resulting from the existence of high relative humidities in cold weather. Water dripping from a ceiling may cause irreparable damage to manufactured articles and machinery. It often results in short circuiting of electric power and lighting systems, necessitating shutdowns and incurring costly repairs. Condensation also causes rotting of wood roof structures, corrosion of metal roofs, and spalling and disintegration of gypsum and other types of roof decks not properly protected. There has been much discussion of this subject in the last few years, but to the writer's knowledge, no general formulae have been advanced for the solution of problems of this nature. Charts of various types have been prepared, but these have been of limited value in most cases because they have applied to specific products.

There are two general types of condensation problems, namely (1) those involving the maintenance of high humidities *necessitated* by manufacturing processes, and (2) those involving high humidities which are the *result* of manufacturing processes. In the former case, the relative humidity is usually controlled and has a definitely fixed value. Buildings *requiring* high humidities include textile mills, tobacco curing rooms, film laboratories, bakery dough rooms, match and candy factories, and printing plants. Where the presence of excessive quantities of moisture is the result of some manufacturing process, the relative humidity and temperature usually vary considerably. Buildings involving this condition include paper mills, laundries, macaroni and canning factories, tanneries, stone and marble plants and cleaning and dyeing establishments.

Condensation is, of course, caused by contact of the warm humid air in a building with surfaces below the dew-point temperature. To prevent condensation it is necessary to do one of two things (1) to increase the temperature of the surfaces upon which the precipitation of moisture takes place above the dew-point for the dry bulb temperature and relative humidity involved, or (2) to reduce the humidity of the air so that the dew-point is below the temperature of the surfaces with which it comes in contact. The fundamental upon which the solution of condensation problems is based is that the drop in temperature through any construction is proportional to the heat resistance.

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Derivation of Formula

In the derivation of a formula for solving condensation problems, the object is to determine the resistance R which a wall or roof must have to maintain the interior surface having a resistance R_s above the dew-point temperature t_d for the dry bulb temperature t and relative humidity r.h. existing in the building, when the outside temperature is t_o. (See Fig. 1). If t_s is the temperature of the interior surface, the limiting value of t_s is t_d, since to prevent condensation the temperature of this surface must not fall below the saturation point or dew-point temperature for the conditions involved.

In order to establish a relationship between R and the other variables entering into the problem, it is assumed that the temperature drop through any part of the wall or roof is proportional to the resistance.

Therefore—

$$\frac{R}{R_s} = \frac{t - t_o}{t - t_s} \quad \text{and}$$

$$R = \frac{R_s\,(t - t_o)}{t - t_s} \qquad (A)$$

Formula A can be used for solving almost any condensation problem by making $t_d = t_s$, if the necessary data are available. From an inspection of the formula it will be noted that condensation can be prevented in three ways, namely:

(1) By decreasing the surface resistance sufficiently to increase t_s to t_d;

(2) By decreasing the relative humidity so that t_d is equal to or less than t_s, and

(3) By increasing the resistance R so that t_s is increased to t_d.

The surface resistance

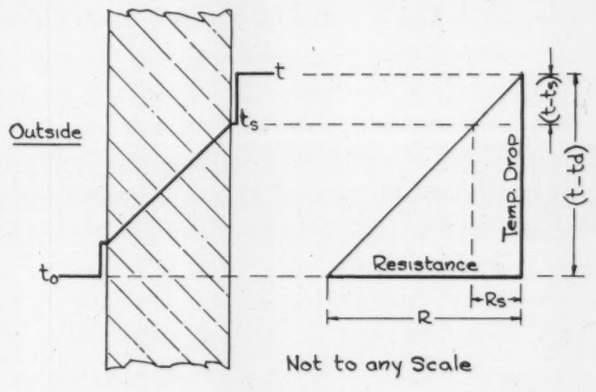

FIG. 1. TEMPERATURE GRADIENT THROUGH WALL WITH DIAGRAM SHOWING RELATION BETWEEN TEMPERATURE DROP AND RESISTANCE

can be decreased by increasing the velocity of air passing over the surface by a fan, blower or other means. This method is commonly resorted to by store merchants for preventing frost or condensation on store windows, by directing fans against the interior surfaces of the windows.

The second method of preventing condensation, namely, that of decreasing the relative humidity, can, of course, be accomplished by dehumidification.

The third method, that of increasing the resistance of the wall or roof, is accomplished by adding a sufficient thickness of insulation to increase the temperature of the interior surface from t_s to t_d.

If, in formula (A)—

$$t_d = t_s$$

$$\frac{x}{k} + \frac{1}{U} = R$$

$$f = \frac{1}{R_s}$$

where—

x = thickness, in inches, of insulation required to prevent condensation.

k = conductivity of insulation in B.t.u. per hour per square foot per degree Fahrenheit per inch thickness.

U = coefficient of transmission of uninsulated wall or roof in B.t.u. per hour per square foot per degree Fahrenheit.

f = conductance of interior surface of wall or roof in B.t.u. per hour per square foot per degree Fahrenheit, then

$$\frac{x}{k} + \frac{1}{U} = \frac{t - t_o}{f\ (t - t_d)}, \text{ and}$$

$$x = k \left[\frac{t - t_o}{f\ (t - t_d)} - \frac{1}{U} \right] \qquad (B)$$

The following example will illustrate the use of this formula:

Determine the thickness of insulation required to prevent ceiling condensation on a roof constructed of 1 in. yellow pine sheathing covered with built-up roofing for an inside temperature at the ceiling of 85 deg. fahr., a relative humidity of 70 per cent, and an outside temperature of —10 deg. fahr., assuming the conductivity of an insulation to be 0.30 B.t.u. per hour per square foot per degree fahrenheit per inch thickness.

k = 0.30 B.t.u. per hour per square foot per degree fahr. per inch thickness.

t = 85 deg. fahr.

t_o = —10 deg. fahr.

r.h. = 70 per cent.

t_d = 74 deg. fahr.

f = 1.34 B.t.u. per hour per square foot per degree fahr. (average).

U = 0.485 B.t.u. per hour per square foot per degree fahrenheit (Table 11A, Chap. I, The A. S. H. & V. E. Guide, 1929).

$$x = 0.30 \left[\frac{85 - (-10)}{1.34\ (85 - 74)} - \frac{1}{0.485} \right]$$

$$= 1.3 \text{ in.}$$

Condensation Chart

In the preparation of the chart (Fig. 2) for ascertaining the thickness of insulation required to prevent condensation, the relative humidity, the inside and outside temperatures, the transmission coefficient of the wall or roof, and the conductivity of the insulation have been considered as variables. The surface coefficient has been assumed to be a constant for still air with a value of 1.34 B.t.u. per hour per square foot per degree fahrenheit.

The procedure for using this chart is as follows:

1. Locate the dry bulb temperature t of the air near the ceiling on scale A, and pass horizontally to the proper relative humidity curve indicated on scale B.
2. From this point pass vertically downward.
3. Locate the difference between the temperature of the air at the ceiling and the lowest outside temperature on scale D, and draw a line horizontally until it intersects line 2. (The vertical line drawn as per paragraph 2.)
4. From the intersection of lines 2 and 3 draw a line to the point P in the lower left-hand corner of the chart.
5. From the intersection of line 4 and line AB, draw a line horizontally until it intersects the diagonal line corresponding to the heat transmission coefficient of the uninsulated roof shown on scale F. (See Chap. I, The A. S. H. V. E. Guide, 1929.)
6. From the intersection found as per paragraph 5, draw a line vertically downward.
7. Locate the conductivity of the insulation to be used (expressed in B.t.u. per hour per square foot per degree fahrenheit) on scale G and draw a line to point Q.
8. From the intersection of lines 6 and 7, draw a line horizontally to the left, and the correct thickness of insulation to use will be indicated on scale H.

Although this chart is intended primarily for roofs, it can be used for walls by using the dry bulb temperature and the corresponding relative humidity of the air near the walls at the point which will necessitate the maximum heat resistance to prevent condensation instead of using the temperature and humidity near the ceiling. If the insulation is installed in such a manner as to alter the number of air spaces in the construction, then the total heat resistance required should be determined from scale E from which the heat resistance of the uninsulated wall should be subtracted to ascertain the resistance to be added to prevent condensation. In this way any increase or decrease in the number of air spaces can be taken into consideration by adding or subtracting the resistance of the air spaces to or from the total resistance to be added by the insulation itself. The thickness of the insulation can then be readily determined if its conductivity is known.

Example:

Determine the thickness of insulation required to prevent ceiling condensation for the following conditions:

Dry bulb temperature near ceiling..	85 deg. fahr.
Relative humidity	70 per cent

Lowest outside temperature.....	—10deg. fahr.
Construction of Uninsulated Roof...	1 in. yellow pine sheathing and built-up roofing.
Coefficient of transmission of roof..	0.485
Conductivity of insulation to be used	0.30

The solution of this problem is indicated on the chart (Fig. 2) by the dotted line.

1. Locate the inside dry bulb temperature of 85 deg. on scale *A*, and draw a line horizontally to the 70 per cent relative humidity curve, indicated on scale *B*.

2. Draw line 2 vertically downward from the intersection located as per paragraph 1.

3. Locate on scale *D* the temperature difference of 95 deg. between the ceiling temperature of 85 deg. and the lowest outside temperature of —10 deg., and draw a line horizontally until it intersects with line 2.

4. From the point of intersection of lines 2 and 3, draw a line to the point *P*.

5. From the intersection of lines 4 and *AB*, draw

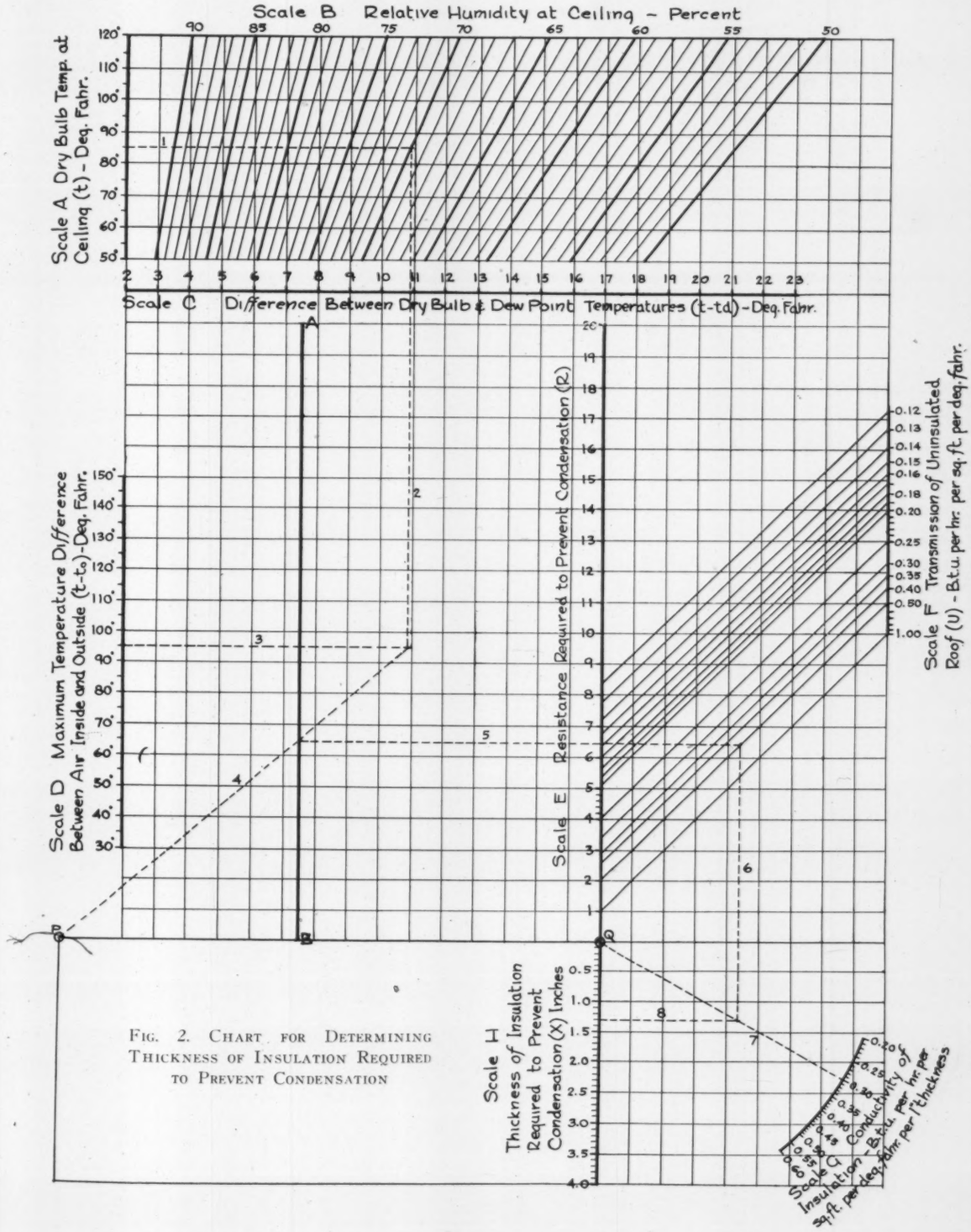
Scale B Relative Humidity at Ceiling - Percent
Scale A Dry Bulb Temp. at Ceiling (t) - Deg. Fahr.
Scale C Difference Between Dry Bulb & Dew Point Temperatures (t-td) - Deg. Fahr.
Scale D Maximum Temperature Difference Between Air Inside and Outside (t-to) - Deg. Fahr.
Scale E Resistance Required to Prevent Condensation (R)
Scale F Transmission of Uninsulated Roof (U) - B.t.u. per hr. per sq. ft. per deg. fahr.
Scale G Conductivity of Insulation - B.t.u. per hr. per sq. ft. per deg. fahr. per 1" thickness
Scale H Thickness of Insulation Required to Prevent Condensation (X) Inches
A
B
P
1
2
3
4
5
6
7
8

FIG. 2. CHART FOR DETERMINING THICKNESS OF INSULATION REQUIRED TO PREVENT CONDENSATION

a line horizontally until it intersects with the diagonal line corresponding to a coefficient of transmission of the roof of 0.485, located on scale *F*.

6. From the intersection found as per paragraph 5, draw line 6 vertically downward.

7. Locate the conductivity of 0.30 B.t.u. per hour per square foot per deg. fahr. of the insulation on scale *G* and draw a line to point *Q*.

8. From the intersection of lines 6 and 7, draw a line horizontally to scale *H*, on which the thickness of insulation of this conductivity is indicated, which is 1.3 in. The nearest commercial thickness above 1.3 in. would, of course, be selected.

Preventing Condensation for High Humidities

Referring to formula *B*, it will be noted that as the relative humidity increases and approaches 100 per cent, the dew-point temperature approaches the dry bulb temperature, the quantity $(t - t_d)$ decreases and approaches zero, and the value of x approaches infinity. Theoretically, an infinite thickness of insulation is required when the relative humidity is 100 per cent and the outside temperature is lower than the inside temperature and the quantity $(t - t_o)$ therefore has a positive value.

As the relative humidity increases beyond a certain point, the thickness of insulation increases very rapidly, and a practical limit to the use of insulation is reached. A thickness of 10, 15 or 20 in. of insulation would be out of the question in most cases. The limiting value of the relative humidity for which condensation can be corrected solely by insulation is usually about 90 per cent, but of course, depends on a number of factors, including the construction, the temperature conditions and the type of insulation.

Humidities of 90 per cent or higher are seldom encountered in any manufacturing establishment, because it is usually difficult to approach saturation to this degree under practical operating conditions. Furthermore, humidities this high are considered very unhealthful. However, in instances where such conditions are maintained and condensation is to be prevented, it can be accomplished in several ways. One method is to install a ceiling a suitable distance from the underside of the roof and to force dry heated air into the plenum space thus created between the ceiling and the roof deck to maintain the temperature of the under surface of the ceiling above the dew-point temperature. Under the circumstances, the roof should be well insulated as otherwise there will be an excessive loss of heat through the roof structure. Another method is to blow dry heated air against the ceiling, but this is usually an extravagant and costly process, not only from the standpoint of first cost, but also from the standpoint of maintenance. Moreover, the results are not always satisfactory as it is usually difficult to obtain the proper distribution of the dry heated air over the ceiling surfaces. The first method is usually to be preferred.

Where Should Insulation Be Applied?

From the theoretical standpoint, the most effective results are obtained by applying the insulation to the interior surface of the wall or roof, or as near in the wall or roof to the interior surface as possible, especially if the building is allowed to cool at night and is heated quickly in the morning. The reasons for this will be evident from the following postulates:

(1) The materials nearest the interior surface will have the greatest increase in temperature for any given increase in the inside temperature, if the outside temperature remains constant.

(2) The product of the specific heat, density and volume of the insulation in a given section of the wall or roof is usually less than that of the brick, concrete, stone, tile or other structural materials in the same section.

(3) The absorption of a given quantity of heat by the less dense insulation will raise the temperature of the insulation more than will the absorption of the same amount of heat raise the temperature of the more dense structural materials of the wall or roof.

(4) An insulation will therefore usually increase in temperature more rapidly at a given rate of heat absorption than will the other materials used in the construction.

(5) Consequently, if the insulation is installed on the interior of the wall or roof, there will be less lag between the room temperature and the interior surface temperature during the heating-up period before the temperature of the wall has reached a state of equilibrium.

There are other factors, however, of perhaps even greater importance than the foregoing, which make it advisable to apply the insulation as far from the interior surface of the wall or roof as possible. Probably the most important is that of providing the necessary vapor protection to the insulation, for no insulation will function satisfactorily if it is not properly vaporproofed. All commercial insulations are more or less porous and without adequate surface protection, moisture laden air will penetrate the insulation until the dew-point temperature within the insulation is reached, at which point condensation will take place.

Bituminous emulsions and paints have been used to some extent to protect insulations applied to the interior surfaces of walls and roofs for the prevention of condensation, but in most cases these have been found wanting. The main reason perhaps is that the irregularities and projections of the surface of the insulation make it difficult to obtain an unbroken seal. Cement plasters have been used with some success for protecting the insulation, but these too have failed in many cases where severe conditions have been involved, due primarily to the tendency of the plaster to crack as is, of course, true of practically all plasters.

Sheet metal will provide a vapor tight seal if the joints are properly soldered, but certain practical difficulties lessen the value of this material for the purpose. The writer's experience has been that the most satisfactory method of vaporproofing a wall or roof is by means of the membrane system consisting of alternate layers of saturated roofing felt and bitu-

men applied between the warm humid air of the building and the insulation. Practically all of the insulated roofs of the textile mills of the south are protected in this manner, that is, by the application to the roof deck of several plies of roofing felt and pitch or asphalt before the insulation is "mopped in." Whether the roof deck be of concrete, wood, steel, tile or gypsum, the insulation should be mopped and not nailed to the deck through the roofing felt, if the temperature and humidity conditions are severe, for it is obvious that nails will puncture the vaporproofing course and defeat the purpose for which it is intended. Needless to say, the outside surface of the insulation must be protected from the rain and snow by means of built-up roofing in the case of roofs, and by other suitable methods in the case of walls.

It is frequently the practice to attempt to correct condensation by installing one or more thicknesses of a given type of insulation to the underside of the roof deck, particularly where the roof deck is of wood and the roof rafters thus provide an easy means of attachment. The thickness of the insulation may be sufficient, and the surface of the insulation adequately protected, but in most cases the vapor will penetrate the joints of the insulation even if they are caulked with a suitable plastic, and condensation will take place on the underside of the roof deck, and in turn drop down upon the insulation, rendering it valueless in due course of time. As previously stated, the most satisfactory results are usually obtained where the insulation is applied over a membrane vaporproofing course on top of the roof deck, or similarly protected if applied to walls.

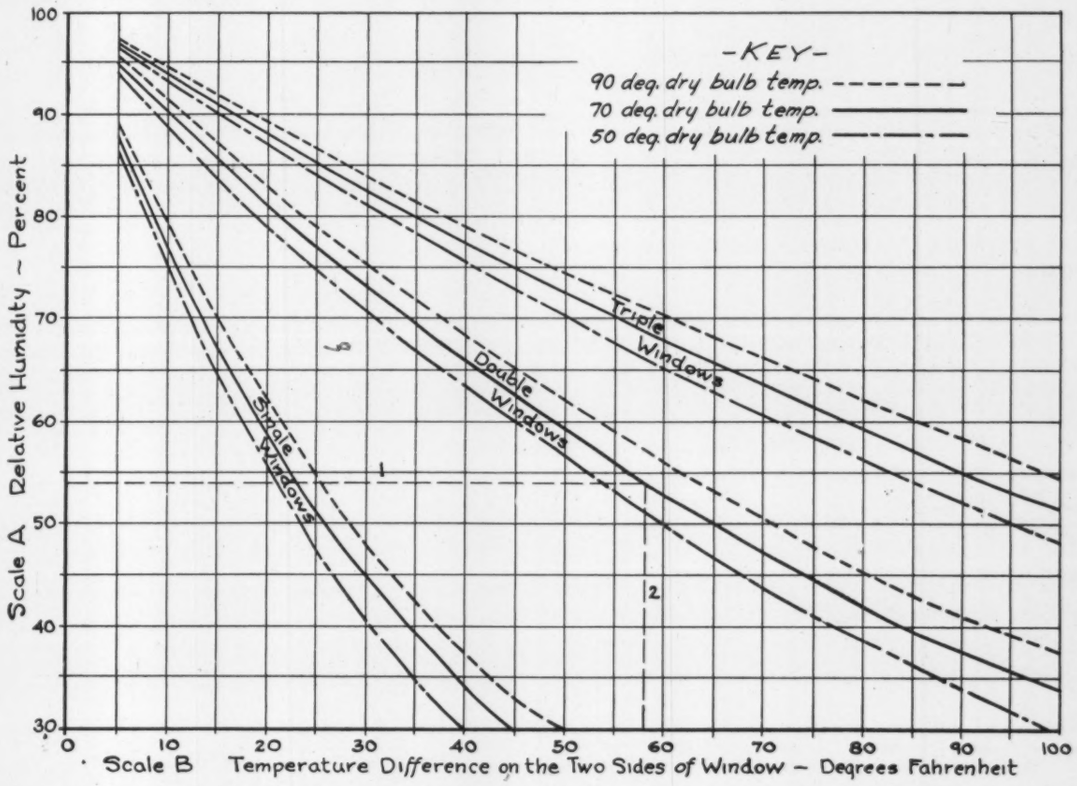

FIG. 3. SATURATION CURVES SHOWING RELATIONSHIP BETWEEN RELATIVE HUMIDITY AND TEMPERATURE DIFFERENCE OR HEAT HEAD FOR SINGLE, DOUBLE AND TRIPLE WINDOWS

Moisture Deposits Not Always Condensation

The deposition of moisture on the interior surfaces of buildings due to other causes is frequently attributed to condensation. For example, the formation of moisture on brick, tile and concrete and other masonry walls, often thought to be due to condensation, is quite often caused by the porosity and inherent capacity of such walls for retaining or transmitting moisture. Roof leaks, too, are often mistaken for condensation deposits. In many cases it is difficult to determine whether the precipitation of moisture is due to condensation or to some other cause.

Condensation on Windows

An interesting paper entitled Frost and Condensation on Windows by L. W. Leonhard and J. A. Grant, was submitted at the Annual Meeting of the A. S. H. & V. E., Chicago, January 28-31, 1929. This paper was based on experimental work done in the Mechanical Engineering Laboratory of the University of Michigan to "determine quantitative values for some of the principal factors that influence the inside temperature of windows."

Fundamentally, the problem of preventing condensation on windows is no different than that of preventing condensation on walls or ceilings. Any of the three methods outlined on page 417 can be used to prevent or reduce condensation on windows. However, it is not economical to decrease the inside surface resistance as per method (1), since the two surface resistances of a single pane of glass comprise almost the entire resistance of the glass, and therefore any reduction in the resistance of either of the

surfaces will result in an appreciable reduction in the overall resistance, and a corresponding increase in the overall transmission. Method (2) involving dehumidification, cannot be resorted to where the high humidity is a necessary part of the manufacturing process. Furthermore, it may not be feasible from the standpoint of the cost of installation and operation of the air conditioning apparatus to attempt to reduce the humidity of the air in the building. It is therefore necessary to resort to method (3) in many instances where it is desirable that window condensation be eliminated, which involves the process of increasing the overall resistance R of the glass. However, the internal heat resistance of glass is practically nil, so that the only means of appreciably increasing the overall heat resistance is by increasing the number of air spaces.

Window Condensation Chart

The chart (Fig. 3) is intended to be used for determining the number of panes of glass required to prevent condensation for certain temperature and humidity conditions, or for determining the outside temperature at which condensation will take place on the inside surface of single, double or triple glass for the humidity and temperature conditions involved. This chart is based on formula (A), assuming still air on the inside glass surface, and a wind exposure of 15 miles per hour on the outside glass surface, using the overall transmission coefficient for glass given in Table 13-A, Chapter I, THE GUIDE, 1929, and the still air surface coefficient for glass of 1.50 B.t.u. per hour per square foot per degree difference in temperature given in Table 4 (Ibid).

The operation of this chart is simple. To determine the type of glass required to prevent condensation, locate the relative humidity on scale A and the difference in temperature between the air on the two sides of the glass on scale B, and the curve corresponding to the inside temperature conditions immediately above the intersection of the lines drawn from these two scales indicates the type of glass required. For example, if the relative humidity is 60 per cent (scale A) and the inside and outside temperatures are 70 deg. and 0 deg., respectively, the temperature difference (scale B) will be 70 deg., and triple pane glass will therefore be required to prevent condensation as indicated by the curve immediately above the intersection of the lines drawn from scales A and B. It will be noted that curves are given for dry bulb temperatures of 50, 70 and 90 deg., respectively. It is apparent that the difference between the three dry bulb temperatures is small.

On account of the fact that the transmission of even triple glass is relatively high and, hence, the resistance correspondingly small, the humidity that can be carried without condensation taking place is low, and the greater the difference between the inside and outside temperatures, the lower the humidity that can exist in the building without condensation taking place on the windows. In some cases condensation on vertical surfaces such as walls and windows is not regarded as a serious objection. For this reason, and also because of the impossibility of entirely preventing condensation under severe conditions and the difficulty of hermetically sealing the space between the panes of glass, no effort is made in such cases to reduce or prevent the precipitation of moisture on windows. It is usually the practice, under the circumstances, to provide gutters under the windows to drain the condensation.

The window condensation chart can also be used for determining the outside temperature at which condensation will take place for any given type of glass, and for any specified inside temperature and humidity conditions. If, for example, double windows are installed in a building, and the inside temperature and humidity are 70 deg. and 54 per cent, respectively, condensation will take place when the temperature difference is 58 deg., or the outside temperature is 70-58 or 12 deg. fahr. The dotted lines (marked 1 and 2) on the chart indicate the solution of this problem.

Conclusion

Condensation on the interior surfaces of buildings is often a serious problem, and one which the heating engineer is frequently called upon to solve. It is caused by the contact of the warm humid air in a building with surfaces below the dew-point temperature, and can be remedied by increasing the temperature of such surfaces above the dew-point temperature or by lowering the humidity.

Dehumidification is not permissible in many cases on account of the fact that a high moisture condition is necessary for manufacturing processes. Hence, the only alternative is to increase the surface temperature, which can be accomplished by decreasing the surface or filament resistance by increasing the velocity of air passing over the surface, or by increasing the overall resistance by adding a sufficient thickness of insulation.

The latter method is generally resorted to, and the thickness of insulation is determined by ascertaining the amount of resistance to be added to increase the temperature of the interior surface above the dew-point temperature for the maximum conditions involved, which in turn is based on the fundamental principle that the drop in temperature is proportional to the resistance.

In order to function satisfactorily the surface of the insulation must be adequately protected from the moisture laden air by means of a suitable vapor-proofing course, preferably one consisting of alternate layers of saturated roofing felt and bitumen. It is usually advisable to install the insulation as far from the interior surface of the wall as possible.

Condensation on windows is most readily prevented by using multiple panes of glass with air spaces between, but the heat resistance provided by each air space is relatively small compared with most insulations, so that usually several air spaces are required to entirely prevent condensation under severe conditions. Condensation on vertical surfaces such as walls and windows, especially the latter, is often disregarded, since the most annoying and troublesome problem is usually that of roof or ceiling condensation, to which more attention is paid.

Errors in the Measurement of the Temperature of Flue Gases[1]

By P. Nicholls[2] (Member) and W. E. Rice[3] (Non-Member), Pittsburgh, Pa.

THE determination of the temperature of the flue gas is an essential operation in all tests of boilers, and it is important that the measurements should be accurate. Errors may arise from a number of causes which can be broadly divided into: (1) Errors due to inaccuracies of the thermometer or instruments used with it, (2) errors due to the failure of the thermometer to indicate the true temperature of the gas stream in contact with it, and (3) errors due to the fact that the temperature as measured may not represent the true average of the whole gas stream.

Errors due to inaccuracies of the temperature measuring instruments used will not be discussed. It is common practice, and it is also most convenient, to use thermocouples in conjunction with a millivoltmeter or a potentiometer—the latter by preference. With reasonably cheap instruments the error can be kept within plus or minus 5 degrees.[4] Too much dependence should not be placed on one couple; it is advisable to have two holes in the flue pipe, one for the couple being used and the other for the insertion of another couple to ascertain whether the first has maintained its calibration. The permanent insertion of two couples, connected to the instrument by a double-throw switch, is a safe practice, since it rarely happens that both couples go bad at the same time.

The object of this paper is to discuss the second source of errors—namely, those due to the failure of the couple to register the true temperature of the gas stream in contact with it. Statements were made during the January, 1927, meeting of the Society, at conferences and in the discussions of the Codes for the Testing and Rating of Low Pressure Boilers, that the flue gas temperature measured in the way required by the Code for Testing might differ as much as 200 to 250 degrees from the true temperature of the gas because of the exposure of the thermocouple to surfaces at temperatures different to those of the gas. The committee which prepared those codes had recognized that there would be some error due to this cause and had pointed this out in an earlier report; and, furthermore, they had suggested that the instructions for testing were rigid enough to insure that the error would be approximately the same for boilers of the same output.

Fig. 1 will be used to illustrate the principles underlying the measurement of temperature by an exposed thermocouple. A couple in the center of the flue at *A* will take heat from the gases flowing by because of its contact with them; but, because it is exposed to the metal of the boiler through the spherical angle or cone *aAa'*, it will radiate heat to these colder surfaces. The surfaces of the flue pipe will also be at a lower temperature than the gases, although this difference will be reduced by the insulation of the flue; the couple will, therefore, be radiating heat to the flue also. As the couple is gaining heat by conduction from the gases and losing it by radiation to the cooler surfaces, it will take a temperature somewhere between those of the gas and those of the surfaces.

A couple at *B* is subjected to the same conditions except that the gases, in passing from *A* to *B*, will have lost some heat by conduction through the insulation. Its exposure to the boiler's surface is represented by spherical angle *bBb'*, which is smaller than that for the couple at *A*, and consequently its loss of heat by radiation to the boiler surfaces will be less than that of *A*, although this decrease is counteracted somewhat by a large exposure to the surface of the flue.

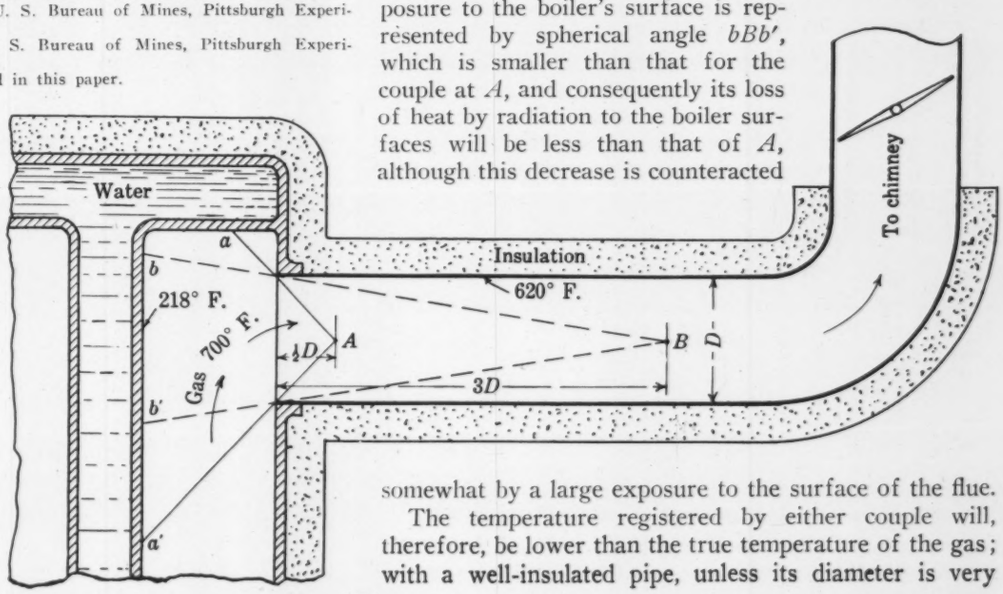

FIG. 1. DIAGRAMMATIC CROSS SECTION OF SMOKE HOOD AND FLUE PIPE OF A HEATING BOILER

The temperature registered by either couple will, therefore, be lower than the true temperature of the gas; with a well-insulated pipe, unless its diameter is very

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Paper presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Bigwin Inn, Lake-of-Bays, Ontario, Canada, June, 1929.

[1] Published by permission of the Director, U. S. Bureau of Mines.

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[4] The Fahrenheit scale is used in this paper.

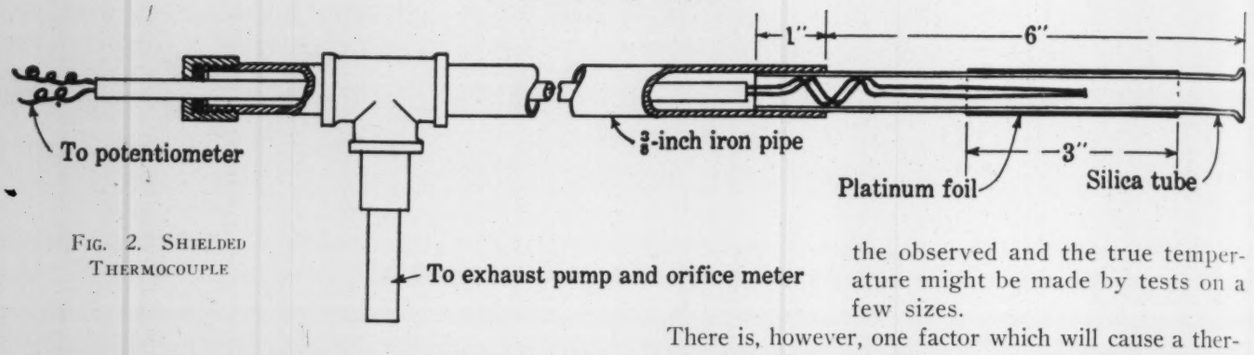


FIG. 2. SHIELDED THERMOCOUPLE

small, one would expect the temperature of B to be greater than that of A, and that B would more nearly indicate the true temperature of the gas.

It will be noted that the relative loss of heat by radiation of couples at A and B is independent of the diameter of the flue provided the distances of A and B from the end of the pipe are the same proportions of the pipe diameter, $\frac{1}{2}D$ and $3D$ in the figure. The relative temperatures registered by A and B will therefore also be independent of the diameter of the pipe except for the effect on the gas temperature of the loss of heat through the insulation; the proportionate amount of heat lost this way for a fixed velocity of gas in the flue will increase with decrease in pipe diameter; that is, the gases will have a larger drop in temperature as they travel along the flue as its diameter decreases. This loss of heat can be computed with fair accuracy and it is therefore not necessary to experiment with a range of boiler sizes and pipe diameters; safe deductions on the relation between the observed and the true temperature might be made by tests on a few sizes.

There is, however, one factor which will cause a thermocouple in the center of a large-diameter flue to depart less from the true temperature of the gas in contact with it than one in a small flue: Heat radiations pass through air without being absorbed, but carbon dioxide and water vapor have an appreciable absorptive power, so that they will shield the couple; this shielding will increase with the thickness of the layer of gas—that is, with the radius of the flue.

TABLE 1.—COMPARISON OF TEMPERATURES, DEG. FAHR., MEASURED BY EXPOSED AND SHIELDED THERMOCOUPLES IN SET-UP SHOWN IN FIG. 3

APPROXIMATE TEMPERATURE	EXPOSED THERMOCOUPLES					SHIELDED THERMOCOUPLE BETWEEN 3 AND 4	METAL SURFACE	APPROXIMATE ERROR OF EXPOSED THERMOCOUPLE
	1	2	3	4	5			
400	405	404	397	393	393	400	353	5
700	717	714	710	698	696	714	588	10
800	830	826	816	819	809	827	713	10
900	915	907	899	899	890	909	787	10
1000	1002	996	989	986	979	1000	875	12

Tests were made on four existing set-ups of boilers. No attempt was made to obtain scientific data to connect the results of the tests with computations; the work was confined to measurements under conditions conforming to boiler test practice. Several thermocouples were placed along the flue, their junctions in all cases being on its center line; an additional couple was peened into the metal of the flue to register its temperature. The thermocouples were all of No. 22 gauge (0.0253 in.) and were calibrated before the tests.

The true temperature of the gas was measured at one place by what is usually termed the velocity method. Fig. 2 shows the device used; its principle consists in the shielding of the couple and the drawing of the gases over it at such a high velocity that the loss of heat by radiation is small compared with the gain by conduction from the fast-moving gas; in addition the couple is shielded from direct radiation to the cold surfaces by the silica tube which, further to reduce radiation, had a bright platinum foil wrapped around it. The radiation coefficient of the platinum to iron surface would be about 0.06 as compared with silica to iron surface of 15 times as much. The junction of the couples and a length of wire from it were kept central and free from contact with the silica tube by forming a spiral of the wires as shown in the figure. Porcelain supports in the silica tube would be more convenient.

Fig. 3 shows the first set-up tested; it consisted of a

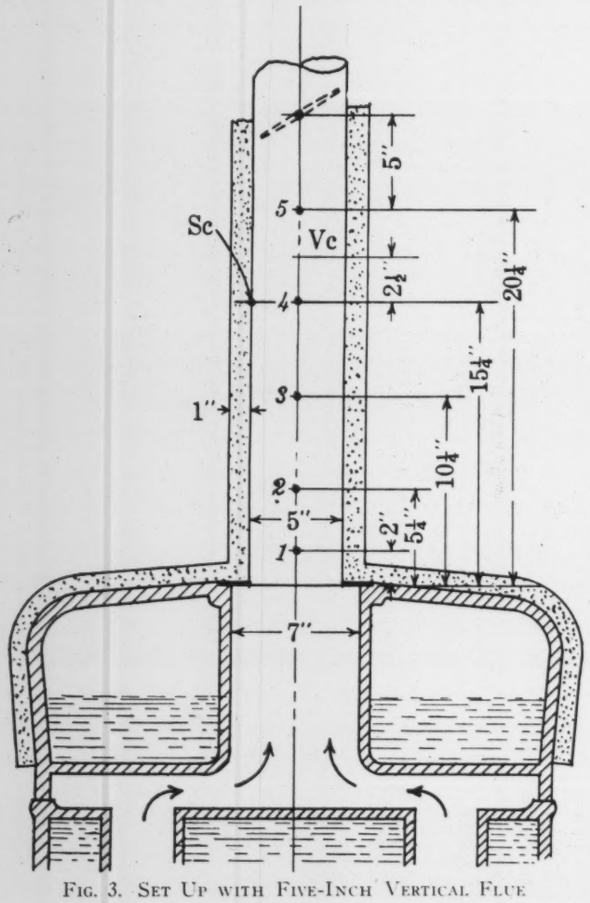


FIG. 3. SET UP WITH FIVE-INCH VERTICAL FLUE

5-in.-diameter vertical flue pipe attached to a vertical sectional boiler. The positions of the exposed couples are indicated by the numbers. The shielded velocity couple is designated as *Cv*, and the surface couple as *Cs*. Observations were made at rates of burning giving flue gas temperatures of approximately 400, 700, 800, 900 and 1,000 degrees. One of the couples was attached to a recording potentiometer so that the temperature of the flue gas could be followed; when the temperature had remained constant at a desired value for at least 10 minutes, several series of readings of the couples were taken and the values for each couple were averaged in order to eliminate errors due to small changes in the temperature of the gases while the readings were being taken.

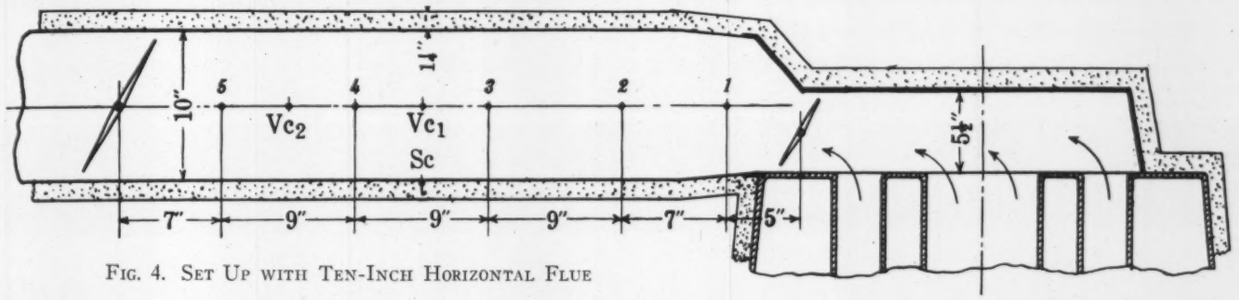


FIG. 4. SET UP WITH TEN-INCH HORIZONTAL FLUE

Table 1 shows the summarized results. The first column designates the series of tests by the approximate flue-gas temperature. The temperatures registered by the exposed couples 1 to 5 show that in all tests, that of couple 1 was the highest in spite of the fact that it had the most exposure to the cold surface of the boiler; also, in all tests the temperature falls as the distance from the boiler increases. This fall in temperature registered by the exposed couples is greater than can be accounted for by the cooling of the gases because of loss of heat to the surface of the flue; for the 800 degree test, the difference between the temperatures of couples 1 and 5 was 21 deg. fahr.; the computed drop in temperature due to loss of heat through the insulation is 9 deg. fahr. It is probable that this lack of agreement is partly due to the center of the gas column as it enters the flue being at a higher temperature than the gas at the circumference which has been in contact with the surfaces of the boiler.

The last column shows the error in the temperatures registered by the exposed couple; these values are the difference between the true temperature and that of an exposed couple located at the same position as the shielded couple. The error has an approximately constant value of 10 deg. fahr. over the range of 700 to 1,000 deg. fahr.

Fig. 4 shows another set-up that was tested. The diameter of the flue pipe in the set-up of Fig. 2 was small for the boiler used, whereas in Fig. 4 it was rather large. Also, the flue pipe was horizontal, had little exposure to cold boiler surfaces, and had a damper which, though wide open, would tend to disturb the flow of the gases.

The tests were conducted in the same manner as previously and Table 2 shows the results.

The values show the same general relations to each other as those in Table 1 but the error of the exposed couple is larger, being about 30 deg. fahr. Some of this increase may be due to the lower velocity of the flue gas which, for similar temperatures, would be under two-thirds of that in the previous tests; also traverses of the flue diameters showed larger variations in temperature and a greater drop as the sides were approached, which would be expected with the horizontal pipe and the damper disturbing the flow.

As a check on the tests with the 10-in. flue the position of the shielded couple was changed from between couples 3 and 4 to between 4 and 5. Table 3 shows the results of these tests.

TABLE 2.—COMPARISON OF TEMPERATURES, DEG. FAHR., MEASURED BY EXPOSED AND SHIELDED THERMOCOUPLES IN SET-UP SHOWN IN FIG. 4

Approximate Temperature	Exposed Thermocouples					Shielded Thermocouple Between 3 and 4	Metal Surface	Approximate Error of Exposed Thermocouple
	1	2	3	4	5			
400	420	418	414	408	408	422	340	10
700	724	719	715	707	706	737	608	26
800	833	822	817	804	803	843	705	33
900	919	916	913	903	902	938	813	30
1000	1013	1010	1009	998	...	1034	910	30

The results are very similar to those of Table 2. There was an error of only 5 deg. fahr. at the 400-degree temperature, as against an error of 10 deg. fahr. in the previous test. This is explained by the higher temperature of the metal pipe, the difference between the true temperature and the metal temperature being 35 degrees in Table 3 and 82 degrees in Table 2,

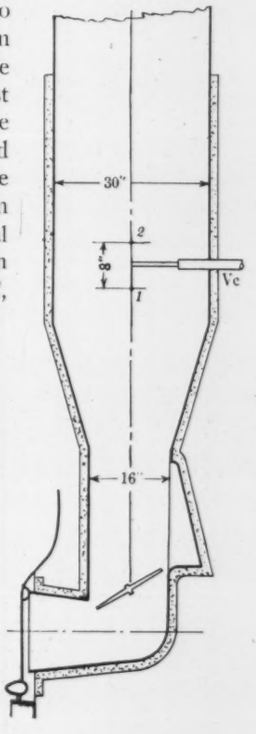


FIG. 5. SET UP WITH THIRTY-INCH VERTICAL FLUE

TABLE 3.—COMPARISON OF TEMPERATURES, DEG. FAHR., MEASURED BY EXPOSED AND SHIELDED THERMOCOUPLES IN SET-UP SHOWN IN FIG. 4

Approximate Temperature	Exposed Thermocouples					Shielded Thermocouple Between 4 and 5	Metal Surface	Approximate Error of Exposed Thermocouple
	1	2	3	4	5			
400	405	401	398	397	394	400	365	5
700	743	737	729	718	717	750	596	32
1000	1001	998	995	988	985	1020	868	34

so that the exposed couple lost less heat by radiation and would consequently register a temperature nearer to the true value.

As a further check on the values with a 10-inch diameter pipe, the set-up of Fig. 4 was varied by using a short horizontal flue and then a vertical leg. Three exposed thermocouples were placed in the vertical flue, 14, 23, and 32 inches from the center line of the horizontal flue, with the shielded couple midway between the last two. Table 4 shows the results obtained.

The computed average drop in temperature along the flue caused by loss of heat through the insulation would be small, although the insulation was only 1 in. thick; in the test at 600 degrees it would be about half a degree in passing from couple 1 to couple 2. The differences registered became less as the temperature increased, that is, as the velocity of the gas was greater, which produced better mixing.

TABLE 4.—COMPARISONS OF TEMPERATURES, DEG. FAHR., MEASURED BY EXPOSED AND SHIELDED THERMOCOUPLES IN A 10-IN. DIAMETER VERTICAL FLUE

Approximate Temperature	Exposed Thermocouples			Shielded Thermocouple Between 2 and 3	Metal Surface	Approximate Error of Exposed Thermocouple
	1	2	3			
400	395	392	391	397	364	5
700	692	690	691	706	614	16
800	793	790	789	805	720	15
900	902	897	895	917	807	21
1000	990	986	990	1014	884	26

The exposed couples gave temperatures 10 to 16 degrees lower than the true gas temperatures and showed greater uniformity of difference than in the tests of the small-diameter flues. Traverses across the flue diameter showed that the temperature did not, on the average, vary more than 2 or 3 degrees within 5 in. on each side of the center of the pipe.

The occasion was also taken to make some measurements in the set-up which was available as shown in Fig. 6. The object of this was to see whether the couple located at position 3 would show higher values than the couple at position 1. Couple 1 is exposed to the cold surfaces of the boiler, whereas couple 3 is entirely shielded from them. No shielded couple was used in these tests.

The errors shown by the last column are of the same order as those with the horizontal flue, but somewhat smaller. It will be noted that in the 800-degree tests the difference between the readings of couples 1 and 3, which were 18 in. apart, was only 4 degrees, whereas the difference was 15 degrees for the same distance in the horizontal flue. The difference in temperature computed from the heat lost through the insulation is about 4½ degrees; this shows that the gases must be stratified as they leave the boiler but are well mixed by the right-angle bend.

Fig. 5 shows the set-up used in the test of a 30-in. diameter vertical flue. The couples had no exposure to the boiler surfaces and only two were used, with the shielded couple midway between them; they were located about 2½ pipe diameters from the start of the flue. The flue was insulated one pipe diameter beyond the last couple. Table 5 shows the results of the test.

Table 6 shows the results. At 400 degrees couple 3 was higher than couple 1 but at all other temperatures it was lower. It would not seem to be of any advantage to measure the temperature in the bend.

Summary and Conclusions

The measurements here reported were all made with an insulated flue pipe; larger departures from the true temperature would have been found with an uninsulated pipe.

In all the set-ups tested the temperature registered by the thermocouple nearest the boiler was higher than those farther away, showing that the stratification of the gases has more influence than has the exposure of the couples to the cooler surfaces.

The temperature registered by the exposed couple in the 5-in. and 30-in. diameter flues departed from the true gas temperature by amounts of the same order—10 to 15 degrees. The differences in the 10-in. diameter horizontal flue were greater, but no proved explanation of this fact can be offered; that it was due to differences in the radiation loss is not probable because the corresponding surface temperatures of the metal in Tables 1 and 2 are approximately the same. The more reasonable explanation is that the stratification of the gases was such that the velocity couple did not draw into its tube a stream of gas representative of that flowing by the exposed couple. There is a possibility, however, that the temperatures as given by the shielded couple in the 5-in. diameter flue are low because the size of the pipe was too small to properly insert the shielded couple device.

The tests on the 30-in. flue give the most reliable data

TABLE 5.—COMPARISONS OF TEMPERATURES, DEG. FAHR., MEASURED BY EXPOSED AND SHIELDED THERMOCOUPLES IN SET-UP SHOWN IN FIG. 5

Approximate Temperature	Exposed Thermocouples		Shielded Thermocouple Berween 1 and 2	Metal Surface	Approximate Error of Exposed Thermocouple
	1	2			
400	395	388	402	348	10
500	475	470	482	406	10
600	596	593	607	520	13
700	712	711	725	576	16

TABLE 6.—TEMPERATURES, DEG. FAHR., MEASURED BY EXPOSED THERMOCOUPLES IN SET-UP SHOWN IN FIG. 6

APPROXIMATE TEMPERATURE	EXPOSED THERMOCOUPLES			
	1	2	3	4
400	405	413	408	394
700	706	690	686	682
800	875	865	861	849
900	941	937	931	915
1000	1033	1034	1026	1015

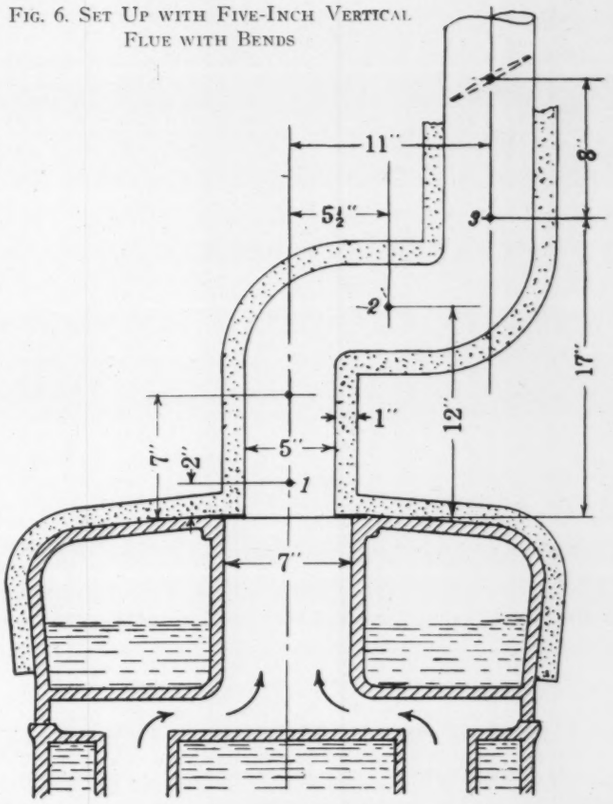
on the departure of the exposed couple from the true temperature of the gas in contact with it, as the shielded couple had a stream of gas at constant temperature to draw from.

Although a thermocouple in the center of the flue indicates a temperature lower than the true temperature of the gas in contact with it, yet this lower value will not differ much from the true average temperature of the gas in the flue because the temperature of the gas in the center of the flue is higher than the average over the cross section.

The tests here reported have therefore shown that the use of an exposed thermocouple installed as specified by the Boiler Testing Code of the Society and with the flue insulated will indicate a temperature which will approach that of the true average temperature of the gases. The probable error should be within 10 degrees.

It is suggested that a standardization of the position of the thermocouple for test purposes should be with its junction on the center line of the flue. The best location would be at 4 equivalent flue diameters from the boiler because this would give time for the gases to mix; this might not always be convenient and it would be satisfactory to allow not less than 2 or more than 4 diameters. The size of the thermocouple wire should not be more than No. 20 *B* and *S* gauge (0.0319 in.).

FIG. 6. SET UP WITH FIVE-INCH VERTICAL FLUE WITH BENDS



The Winter's Coal Supply*

By H. E. Nold, Professor of Mine Engineering, Ohio State University

BUYING the winter's coal supply for the home is a problem of what to buy, when to buy, and, equally important, how to use what is bought.

When to Buy

The summer is undoubtedly the desirable time for purchasing the coal to be used during the winter.

The mines producing domestic coal are seldom forced to work at high pressure during the summer and consequently the coal is often better than that produced during the winter rush.

The retail coal yards are similarly less busy in summer than in winter and can render more satisfactory service.

It is generally true that domestic coal can be bought considerably cheaper in summer than in winter. In fact, the price usually advances about the first of September.

What to Buy

There are three distinct types of coal available for use in Ohio homes. These are anthracite, semi-bituminous, and bituminous, often called "soft coal."

Anthracite, commonly known as "hard coal," is coal which, during its formation, has been subjected to such pressure and heat that almost all the volatile, combustible matter or long flame producing constituents have been driven off and only the "fixed carbon," corresponding roughly to coke, has been left behind.

This coal is mined in eastern Pennsylvania and burns with almost non-luminous flame, making no soot or smoke.

Anthracite, while a desirable fuel to burn, needs special drafting conditions and is high priced so that it can be classed among the luxuries and is, therefore, used only by a few.

Coal of the semi-bituminous or Pocahontas variety, commonly known in Ohio as "smokeless coal," comes from a number of seams in the Pocahontas and New River mining districts of southern West Virginia. This coal is soft, easily broken, and dusty; it is, nevertheless, a high grade fuel and can easily be burned with little

* Engineering Experiment Station News.

smoke and soot. The price, however, is appreciably higher than that of bituminous coals.

The great bulk of house heating in Ohio is done with bituminous coal. This is coal of the long flame variety and is usually hard and blocky.

Bituminous coal is produced from a rather large number of seams distributed over a wide territory. That used in Ohio for domestic heating is mined chiefly in West Virginia, Ohio, Kentucky, and to some extent in Pennsylvania.

There is a mistaken idea among many users that Ohio coal is of low quality and is not well screened and cleaned. Ohio has, and is producing good quality coal, but it is not burned by our citizens to the extent that the quality and price justify.

At the time of peak demand, during and after the World War, a great number of mines were opened in Ohio, many of them inadequately financed and without proper screening and cleaning equipment. Most of these mines have now been abandoned and today, while we have fewer mines, we still have ample production capacity from mines properly equipped to furnish a well prepared product.

The following is taken from the code Relating to Prevention of Smoke in the City of Columbus, Ohio.

How to Use Soft (Bituminous) Coal

A wise man is one who makes the best of what he has. With anthracite practically unknown in our territory and Pocahontas becoming rarer each year, the people in this section of the country must learn to make the best of what fuel they have—the soft (bituminous) coal of the Hocking and Cambridge districts of Ohio as well as those from Kentucky and West Virginia. A little self-education in the right method of "firing," some minor mechanical changes in your present furnace, or an insistence that the new one be properly equipped—and this cheaper, ever-abundant coal will give all of the results heretofore obtained from rarer, more expensive fuels.

There are two types of furnaces for residence heating which are constructed to burn bituminous coal economically and smokelessly, viz: the "underfeed" and the "overfeed."

In the underfeed type, the fresh charge of coal is pushed by mechanical means up into the firepot from underneath, thus compelling the volatile gases from the green coal to pass through the incandescent fire on the top of the fuel bed. This process burns the gases, liberating the heat units which are then available for transference to the air circulating through the furnace casing. In the ordinary surface burning, hot-air furnace, the volatile gases are not consumed but pass out, partially burned, as smoke which, condensing on the metal surfaces as soot, insulate the metal, making the use of more coal necessary. The result is incomplete combustion, soot, more coal, larger coal bills.

In the "overfeed" smokeless hot-air furnace, preheated air is admitted to the fire through a perforated ring section between the firepot and radiator, producing smokeless combustion with bituminous coals of the highest volatility.

In regard to the building and operation of a fire in your furnace from soft (bituminous) coals the following rules, if followed, will give the home-owner satisfactory operation:

1. Clean grate and ash pit thoroughly.
2. Close check draft in smoke pipe and open draft door for fuel draft.
3. Cover grate with small lumps of coal to a depth of six inches.
4. Crumple old newspapers and place on top of coal. Cover with kindling wood and light paper.
5. Leave draft on until coal is well ignited.

The above applies to any kind of furnace and it will be readily seen that this method saves coal and prevents smoke, inasmuch as the volatiles in the coal must pass through the hot part of the fuel bed and be utilized for heating, instead of being wasted up the chimney as happens when the fire is built in the ordinary manner by piling the coal on top of the kindling. Don't forget that bituminous coal will ignite almost as readily from the top as it will from the bottom.

The proper way of adding a fresh charge of coal depends on the type of furnace being used. If of the underfeed type, the fresh charge is pushed up into the firepot from below by operating the lever which actuates the piston in the bottom of the coal cylinder which, of course, is filled before building the fire.

The amount of the charge should be regulated according to weather conditions but must not be sufficient to break through the fuel bed and expose green coal.

In the "overfeed type" observe the following:

1. Fire fresh coal on the low side of the fuel bed.
2. Fire next charge on the opposite side of the fuel bed.
3. Never cover whole fuel bed with green coal.

or

1. Work a depression something like a bowl in the center of the fire, moving the hot fire to the side of the fire pot.
2. Heap the fresh coal in the depression, piling it up like an inverted bowl.
3. On the next charge repeat the process.

This method leaves a ring of hot coals around the fresh charge, helping to ignite and consume the volatiles which cause smoke and soot. It is the better method of the two, as it brings the hottest part of the fire against the metal fire pot where it is needed.

By careful attention to the firing, much smoke, soot, and waste with its consequent damage, may be avoided.

Sulphur tests for the determination of sulphates in boiler scale and sulphur in coal can conveniently be made by means of a blow pipe, according to *Power Plant Engineering*. The scale should be mixed with charcoal and soda and heated with a reducing flame produced by holding the tip of the blow pipe just outside of the smoky part of the flame from a candle or bunsen burner.

This reducing flame, having a deficiency of oxygen, takes away oxygen from the compound. If sulphur is present as a sulphate, it is reduced to a sulphide, a lump of which placed on a silver coin and moistened leaves a black stain.

Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems

By F. C. Houghten[1], W. W. Teague[2], W. Ed. Miller[3], and W. P. Yant[4], Pittsburgh, Pa.

MEMBERS

AN UNDERSTANDING of the relation of man to his atmospheric environment is necessary for progress in the art of heating and ventilation. This principle has long been recognized by leaders in this branch of the engineering profession, and was one of the main factors leading up to the establishment of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Since its establishment, the Laboratory, in cooperation with the U. S. Bureau of Mines, has had under investigation as one of its most important lines of research, the relation between temperature, humidity and air motion and feeling of warmth, comfort, health and the physiological reactions experienced by human beings. This study has resulted in the development of information and data of great value to the public in general, as well as to the heating and ventilating engineer, the physiologist and the physician.

As a result of completion of another phase of this investigation, the Laboratory presented to the Society at the Annual Meeting, information concerning the rate of heat production in, and dissipation from the body, both for still and moving air. The complete report as presented will appear in the TRANSACTIONS of the Society. A paper on this study was prepared for the *American Physiological Society* and appears in *AMERICAN JOURNAL OF PHYSIOLOGY*.

In the complete report heat loss from the body is differentiated into loss by radiation and convection or sensible heat loss, and loss by evaporation of moisture or latent heat loss. A number of curves for solution of practical problems in air conditioning and examples in their use are also presented.

To satisfy the processes of life, man takes into his system food consisting largely of oxidizable material, air containing oxygen as its essential constituent, and water. Through the process of metabolism, carbon, hydrogen and other elements in minute quantities, contained in the food, unite with oxygen from the inspired air developing energy for internal and external work and heat for maintaining body temperature.

The chemical reactions for the oxidation of typical classes of food in the body are exothermic or proceed with an evolution of heat. These reactions are stated by MacLeod[5] as:

1. Carbohydrate:
   $$C_6H_{12}O_6+6O_2=6CO_2+6H_2O$$
   (Dextrose)
2. Fat:
   $$C_3H_5(C_{18}H_{33}O_2)_3+80O_2=57CO_2+52H_2O$$
   (Olein)

[5] *Physiology and Biochemistry in Modern Medicine*, C. V. Mosby Co., by J. J. R. MacLeod, 1926, p. 775.

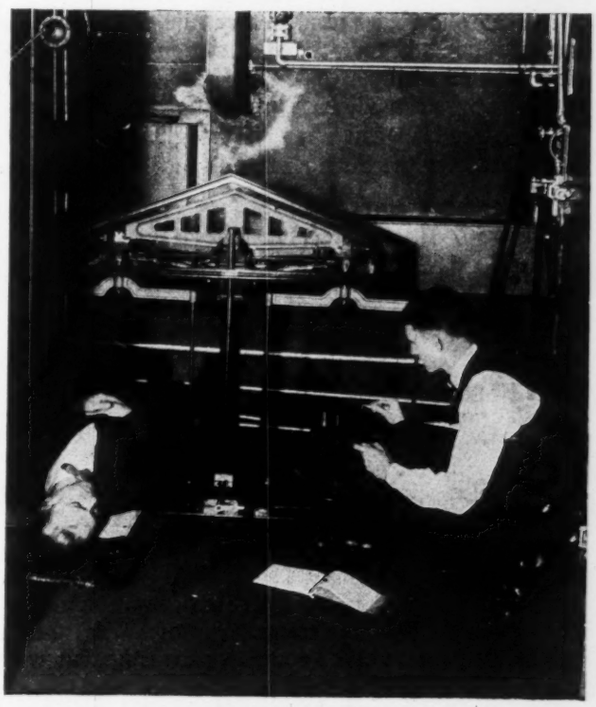


FIG. 1. BALANCE ON WHICH SUBJECTS ARE WEIGHED

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Abstract of paper presented at the 35th Annual Meeting of AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Chicago, Ill., January, 1929.

FIG. 2. METABOLISM APPARATUS FOR DETERMINING RATE AND ANALYSIS OF EXHALED BREATH

3. Protein:

$$C_{72}H_{112}N_{18}O_{22}S+77O_2=63CO_2+38H_2O+9CO(NH_2)_2+SO_3$$

(Albumin)

The calorific values used in determining the heat produced in the body for each of the three classes of foods are those universally accepted by the physiologist. Computations were actually based upon the volume of oxygen consumed rather than upon the weight of food. The values used for the three classes of foods are:

1. Carbohydrate:
   567 B.t.u. per cubic foot of O_2 consumed or 6,789 B.t.u. per pound of dextrose.
2. Fat:
   526 B.t.u. per cubic foot of O_2 consumed or 17,114 B.t.u. per pound of olein.
3. Protein:
   504 B.t.u. per cubic foot of O_2 consumed or 8,646 B.t.u. per pound of albumin.

The object and need of this particular phase of the more general problem of determining the relation between temperature, humidity and air motion and the feeling of warmth, comfort and health of human beings were outlined by the Technical Advisory Committee on this subject, under the chairmanship of W. H. Carrier.[6]

The purpose of the Laboratory investigation was to determine the rate of heat production in the body, the rate of heat dissipation to the air and the differentiation of this loss between that taking place by radiation and convection, or sensible heat loss and loss by evaporation of moisture, or loss of latent heat. It was also the purpose of the investigation to determine the rate of addition of moisture to the air by evaporation from the skin and lungs.

Test Procedure and Observations

In this study 267 tests were made on subjects in the psychrometric chambers of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in the Pittsburgh Experiment Station of the U. S. Bureau of Mines. In either of these two rooms, which are completely described in another report,[7] any desired atmospheric condition may be maintained.

The subjects were university students of good health and average physique between the ages of 19 and 24 years. Since heat production depends largely upon the activity and diet of the subject and since it was desired to have the data apply to persons normally at ease as in an audience hall, their ordinary diet and routine of life was not interfered with outside of the 4½ hours of the preliminary and test periods. During the preliminary and the 4-hour test periods, they sat at a table in a test chamber kept at the desired atmospheric condition. They were normally clothed, and their activity consisted of making a few test observations, and reading, studying or talking as was their pleasure. Unusual or strenuous activity was not permitted, but normal movement of body and limbs was not interfered with.

The data collected during the 4-hour test period constituted a heat balance test not unlike a heat balance test on a boiler. Weighing on the bullion balance, Fig. 1, gave the rate of weight loss of the subject to within plus or minus 0.2 gram. The weight loss is due to evaporation of moisture from the body and to the carbon dioxide exhaled less the oxygen consumed. Measurement and analysis of the exhaled breath with the metabolism apparatus, Fig. 2 gave the rate at which carbon dioxide was produced and oxygen consumed.

The rate of heat production was determined from the analysis of the exhaled breath in accordance with an

[6] Report of the Technical Advisory Committee on Temperature, Humidity and Air Motion, by W. H. Carrier, JOURNAL AMERICAN SOCIETY OF HEATING & VENTILATING ENGINEERS, 1926, vol. 32, pp. 279-280.

[7] Determining Lines of Equal Comfort, by F. C. Houghten and C. P. Yagloglou, Transactions of AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 1923, vol. 29, pp. 163-175.

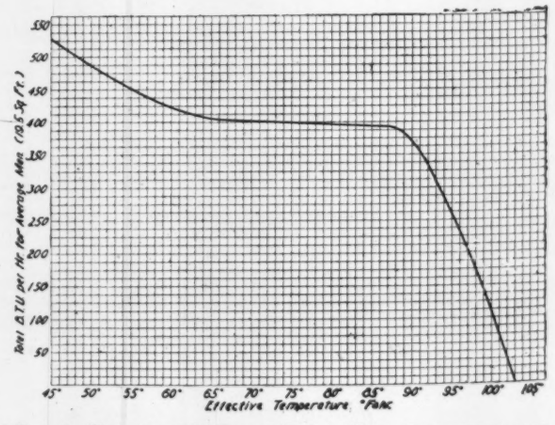

FIG. 3. RELATION BETWEEN TOTAL HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE

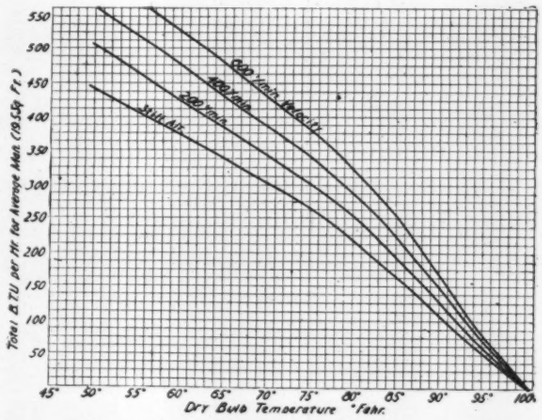

FIG. 4. RELATION BETWEEN SENSIBLE HEAT LOSS FROM HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AND MOVING AIR

accepted method[8] of calculation giving the metabolic rate.

The data collected are for relative humidities of approximately 20, 45, 70 and 95 per cents, effective temperatures ranging from 44 to 100 deg., and conditions of still air and velocities of 235 and 385 ft. per minute. In fixing the limits of the effective temperature range, it was desired to obtain data over the range met within heating and ventilating practice. In order to establish the curves over this range, limits somewhat beyond those dictated by the needs of such practice were chosen. The upper practical limit was fixed at 90 deg. effective temperature. The lower limit was chosen as the lowest which could be endured at rest with the prescribed clothing without undue discomfort.

Beyond the limit of 90 deg. effective temperature four tests were made in order to check the tendency of the curves at higher temperatures. These points show a very definite direction for the curves but may be slightly in error due to the necessarily shorter duration of the test, the inability of the body to adjust itself quickly to abnormal conditions, the excessive temperature rises, the excessive perspiration and the general effect of the hot conditions on the subjects and observers.

Practical Application of Heat and Moisture Loss Data to Air Conditioning Problems

In the solution of such problems the engineer is interested in the total rate of heat dissipation to the atmosphere per average person and in the rate of sensible and latent heat dissipation, and also in the rate at which moisture is added to the atmosphere per person for all atmospheric conditions met with in practice. In the solution of certain problems it is also convenient to know the percentage of the total loss as sensible and latent heat. In Figs. 3, 4, 5, and 6 this information is given in convenient form for practical application. Although total heat loss and sensible and latent heat losses are not exact functions of effective and dry-bulb temperature, respectively, for all conditions of humidity and air motion, they are plotted as such in the curves. This is accomplished by approximations not always rigidly accurate but sufficiently so for application in most practical problems.

The sensible and latent heat loss curves for different air velocities in Figs. 4 and 5 are the result of approximations and extrapolations of the data and may not be rigidly accurate. However, since the total effect of air velocity on these losses is never very great, the approximation shown is probably sufficiently accurate for practical application.

With these precautions the data contained in Figs. 3 to 6 are offered for the use of the engineer in solving most of his practical problems. In some instances, however, in particular cases where extreme accuracy is desired, the variation in the data for different atmospheric conditions must be taken into consideration.

Typical uses of these data are given in the solution of the following problems:

Problem 1-A. How much sensible heat, how much latent heat and how much water vapor will be added per hour to the atmosphere of an auditorium by an audience of 1,000 adults, when the dry- and wet-bulb temperatures are 75 deg. fahr. and 63.5 deg. fahr., respectively?

Problem 1-B. If the dry- and wet-bulb temperature of the auditorium were 85 deg. and 63 deg. respectively, how much heat and moisture would be dissipated to the atmosphere?

Solution. Problem 1-A. From Fig. 4 find the sensible heat loss per person for 75 deg. dry-bulb and still air to be 265 B.t.u. per hour. From Fig. 5 find the latent heat loss per person for 75 deg. dry bulb to be 134 B.t.u. per hour and the moisture added to be 905 grains per hour. 1,000 × 265 = 265,000 B.t.u. sensible heat, 1,000 × 134 = 134,000 B.t.u. latent heat and 1,000 × 905 = 905,000 grains or 129 lb. of water vapor will be added per hour to the air in the auditorium.

These sensible and latent heat loss additions may also be found as follows: The effective temperature of the condition 75 deg. dry-bulb and 63.5 deg. wet bulb is 70.3 deg. effective temperature. From Fig. 3 find 403 B.t.u. as the total heat added to the air by a person for 70.3 deg. effective temperature. From Fig. 6 find the percentages of sensible and latent heat at 75 deg. dry bulb to be 66.5 per cent and 33.5 per cent. The sensible heat

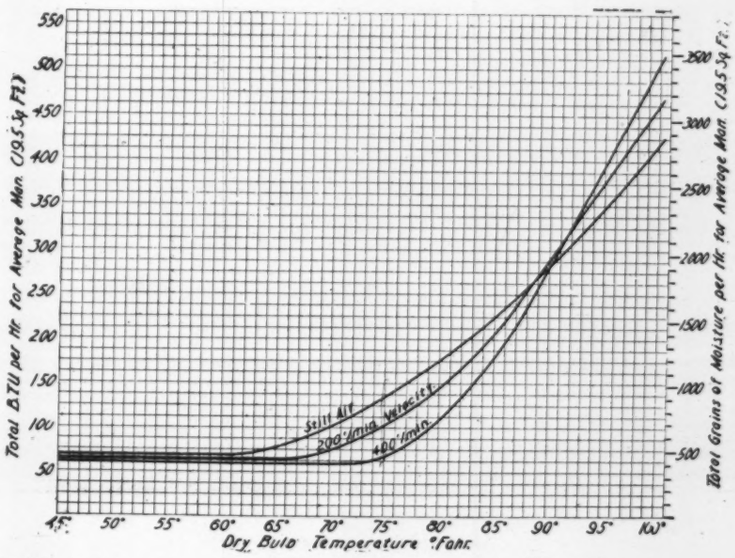

FIG. 5. RELATION BETWEEN HEAT AND WEIGHT LOSS FROM THE HUMAN BODY BY EVAPORATION AND DRY-BULB TEMPERATURE FOR STILL AND MOVING AIR

[8] *Physiology and Biochemistry in Modern Medicine*, C. V. Mosby Co., by J. J. R. MacLeod, 1926, pp. 777-780.

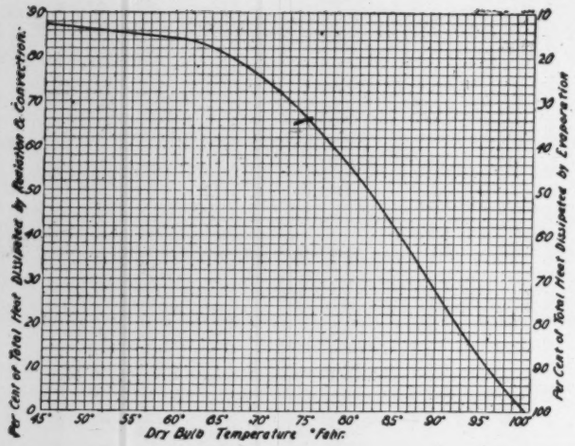
Per Cent of Total Heat Dissipated by Radiation & Convection
Per Cent of Total Heat Dissipated by Evaporation
Dry Bulb Temperature °Fahr.
45° 50° 55° 60° 65° 70° 75° 80° 85° 90° 95° 100°

FIG. 6. RELATION BETWEEN HEAT LOSS FROM THE HUMAN BODY BY EVAPORATION, RADIATION, CONVECTION AND DRY-BULB TEMPERATURE

added to the air in the auditorium is $1,000 \times 0.665 \times 403 = 267,995$ B.t.u. per hour. The latent heat added is $1,000 \times 0.335 \times 403 = 135,005$ B.t.u. per hour.

Solution. Problem 1-B. From Figs. 4 and 5 respectively, the sensible and latent heat losses per person for 85 deg. dry-bulb are found to be 164 and 225 B.t.u. per hour. The water vapor added to atmosphere is 1,520 grains per hour. The audience will then add 164,000 B.t.u. sensible heat, 225,000 B.t.u. latent heat and 1,520,000 grains or 217 lb. of water vapor to the air in the auditorium per hour.

Problem 2. Neglecting the gain or loss of heat to an auditorium by transmission or infiltration through the walls, windows and doors, how many cubic feet of outside air, with 65 deg. dry-bulb, 59 deg. wet bulb and 63.1 deg. effective temperature must be added per hour to an auditorium containing 1,000 people in order that the inside shall not exceed 75 and 65 deg., respectively?

Solution. Problem 2. Figs. 4 and 5 give 265 B.t.u. sensible heat and 905 grains of moisture as the additions per person with 75 deg. dry-bulb in the auditorium. Therefore, 265,000 B.t.u. of sensible heat and 905,000 grains of moisture will be added to the air in the auditorium per hour.

Taking 0.24 as the specific heat of air, 2.4 B.t.u. per pound of air will be required to raise the dry-bulb temperature from 65 to 75 deg. and $\frac{265,000}{2.4} = 110,400$ lb. of air or $110,400 \times 13.4 = 1,479,000$ cu. ft. of air per hour will be required. This is equivalent to $\frac{1,479,000}{1,000 \times 60} = 24.7$ cu. ft. per person per minute.

The moisture content of the inside air as taken from a psychrometric chart is 76 grains per pound of dry air and that of the outside condition is 65 grains so the increase in moisture content will be 11 grains per pound of dry air. $\frac{905,000}{11} = 82,300$ lb. of dry air or approximately 83,000 lb. of air at the specified condition will be required. This is equivalent to $83,000 \times 13.4 = 1,112,000$ cu. ft. of air per hour or $\frac{1,112,000}{1,000 \times 60} = 18.5$ cu. ft. of air per minute per person.

The higher volume of 24.7 cu. ft. per person per minute will be required to keep the dry-bulb from rising above the 75 deg. specified. The wet-bulb will, therefore, not rise to the maximum of 65 deg.

For good ventilation the condition of the air should never be such that sensible perspiration will result. In connection with the collection of the data contained in this report observations were made on the degree of noticeable perspiration experienced by the subjects for different atmospheric conditions. The lowest degree for which 40 per cent of the subjects experienced the conditions of perspiration indicated is given in Table 1 for 20 and 95 per cent relative humidity.

TABLE 1. CONDITION OF SENSIBLE PERSPIRATION FOR VARIOUS ATMOSPHERIC CONDITIONS

	Atmospheric Condition					
	95% R. H.			20% R. H.		
Degree of Perspiration*	E.T.	D.B.	W.B.	E.T.	D.B.	W.B.
Forehead clammy	73.0	73.6	72.4	75.0	87.0	60.7
Body clammy	73.0	73.6	72.4	75.0	87.0	60.7
Body damp	79.0	79.7	78.4	81.0	97.5	67.5
Beads on forehead	80.0	80.8	79.4	87.0	109.4	75.2
Body wet	84.5	85.4	84.0	86.5	108.5	74.6
Perspiration on forehead runs and drips	88.0	89.0	87.6	94.0	125.2	85.4
Perspiration runs down body	88.5	89.5	88.1	90.0	116.0	79.5

* 40 per cent of subjects registered degree of perspiration equal to or greater than indicated.

New National Standards Announced

IMPORTANT developments in the national standardization activities of almost every major American industry are described in the American Standards Year Book, a review of the national industrial standardization movement during the past twelve months just issued by the AMERICAN STANDARDS ASSOCIATION, 29 West 39th St., New York. The review covers mechanical, electrical, building, transportation, mining, textile and many other industries.

Important new standards completed under the auspices of the AMERICAN STANDARDS ASSOCIATION during the past year include: a group of 19 specifications and methods of test for various petroleum products; a comprehensive code for protection against lightning, including sections for protections of persons, of buildings and miscellaneous property, and of structures containing inflammable liquids and gases; standard track gages and car sizes for metal mines; a group of six specifications for bare and insulated copper wire (this is the first time that standard specifications for cotton, silk and enameled magnet wire have been available); two more standards for pipe flanges and fittings; and specifications for track work, covering material for both steam and electric railways.

Important new projects initiated include; specifications for pressure and vacuum gages; specifications for leather belting; specifications for materials and workmanship for plastering; specifications for coal mine cars; and a comprehensive code on mine timbering, including specifications for timber and a code of timbering practice.

The Year Book lists altogether about 150 national standards which have already been completed and about 175 other national projects now under way.

The Year Book also lists nearly 550 trade associations, technical societies and other organizations which have been cooperating in the establishment of standards.

A Simplified Arrangement of the A-A Dust Determinator

By S. G. Sylvan[1], Pittsburgh, Pa.
NON-MEMBER

THE A-A dust determinator is a device for determination of dust quantities suspended in air. The air is drawn to a screen or filter paper, having openings of suitable size and the increase in resistance of the screen is measured. This increase stands in a certain relation to the dust content in the air. In the usual set up, this determinator consists of a resistance gauge, a flow meter, a holder for the screen or filter paper and an air pump with motor and suitable valves for adjustment of the air volume. It can be operated in two different ways: The rate of flow may be kept constant, measuring the variation in resistance, or the resistance may be kept constant, measuring the variations in rate of flow.

Very elaborate tests have been made by Miss Margaret Ingels,[2] using the first way of operation.[3] Miss Ingels arrived at an equation

$$x = 226.76 \left(R_o^{\,0.08} - \frac{R_o^{\,1.26}}{R_1^{\,1.18}} \right)$$

where x is a measurement of the dust accumulation on the filter in an arbitrary unit and R is the resistance of the filter.

In the proposed arrangement, the dust determinator consists of a filter holder, a large bottle and some glass tubing and rubber hose connections. The filter holder consists of two metallic plates, each being drilled for air passage as shown in Fig. 1. One plate is exposed to the air and the other one is adapted to a hose connection. Any number of air passages may be drilled, all except one of which are plugged when operating the determinator. The holder is connected to a tube, extending through a rubber stopper to the bottom of a large bottle (about 1 quart). Another tube also extends to the bottom of the bottle and is connected to a piece of rubber tubing, serving as a siphon. The bottle has three level marks indicating known values, and is filled with water well above the top mark.

When operating the device, water is withdrawn from the bottle by means of the siphon and replaced by air entering through a screen or filter paper, inserted between the two plates. The resistance will be constant all the time, but the rate of withdrawal will decrease with dust accumulation on the filter. The only readings necessary for a determination are the times when the water level passes the three marks.

In order to compute the dustiness of the air from these readings, a mathematical treatment will be necessary. From Miss Ingels' test[4], it is seen that the resistance of filter paper is a linear function of the rate of flow, at least when the velocity is low. The rate of flow is proportional to the area of the filter or, in other words, 2 sq. in. let through twice as much air as 1 sq. in. at the same resistance. This does not quite agree with Miss Ingels' tests probably on account of unevenness in the filter paper tested. Provided that the texture of the paper is uniform, it is mathematically correct. The equation of the resistance curve of the filter paper consequently is

$$c = kR \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

R being the resistance, c the velocity and k a constant, which may have any numerical value depending on the kind of filter paper used.

The volume V of air, flowing through the filter in a certain time t will be

$$V = c\,A\,t$$

where A is the area of the filter. If this area is not constant during the time t, the equation must be written

$$dV = c\,A\,dt \quad \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

The area A is subject to a continuous decrease due to dust accumulation. This decrease is assumed to be proportional to the dusty air volume that has passed through the filter, or, for instance, equal to $n\,V$. Hence

$$A = A_1 - n\,V \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

A_1 being the original area and n a constant, expressing the dustiness of the air. The dimension of this constant is L^{-1}. Elimination of A from (2) and (3) gives

$$\frac{dV}{dt} = c\,(A_1 - n\,V) \quad \ldots\ldots\ldots\ldots (4)$$

which is the general equation for the A-A determinator.

Miss Ingels' tests were made with constant rate of flow i.e.,

$$\frac{dV}{dt} = \text{constant} = C = \frac{V}{t}$$

Substitution of this quantity in (4) and elimination of c from (1) gives

$$C = k\,R\,(A_1 - n\,C\,t)$$

In order to compute n from this equation, two measurements are required; for instance, the resistance at $t = o$ and at $t = t_1$. With $t = o$, the equation gives

$$C = k\,R_o A_1$$

and with $t = t_1$

$$C = k\,R_1\,(A_1 - nCt_1)$$

from which two equations k may be eliminated leaving

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[2]Former research engineer of the A. S. H. V. E. Laboratory, Pittsburgh, Pa.

[3]JOURNAL A. S. H. & V. E. January, 1925.

[4]Reported in the JOURNAL A. S. H. & V. E. February, 1924.

$$n=\frac{A_1}{Ct_1}\left(1-\frac{R_o}{R_1}\right)\ldots.(5)$$

The following dust concentrations were obtained from Miss Ingels' tests using this equation.

TEST NO. 1 AREA 0.0218 SQ. FT. TIME 30 MIN. 0.4 C.F.M.

FILTER NO.	INITIAL RESISTANCE	FINAL RESISTANCE	DUST CONCENTRATION	PER CENT VARIATION
1	4.6	7.8	0.00075	1.4
2	5.8	9.9	0.00073	1.4
3	5.6	9.2	0.00072	2.7
4	5.7	9.8	0.00076	2.7
5	6.2	10.4	0.00073	1.4
6	4.7	7.35	(0.00083)	excluded
		Average	0.00074	

TEST NO. 4 AREA 0.0218 SQ. FT. TIME 30 MIN. 0.4 C.F.M.

FILTER NO.	INITIAL RESISTANCE	FINAL RESISTANCE	DUST CONCENTRATION	PER CENT VARIATION
1	4.65	5.65	0.00032	3
2	5.10	6.2	0.00032	3
3	4.8	5.7	0.00028	0
4	4.4	5.2	0.00028	10
5	4.9	5.9	0.00031	0
6	5.65	6.85	0.00032	3
		Average	0.00031	

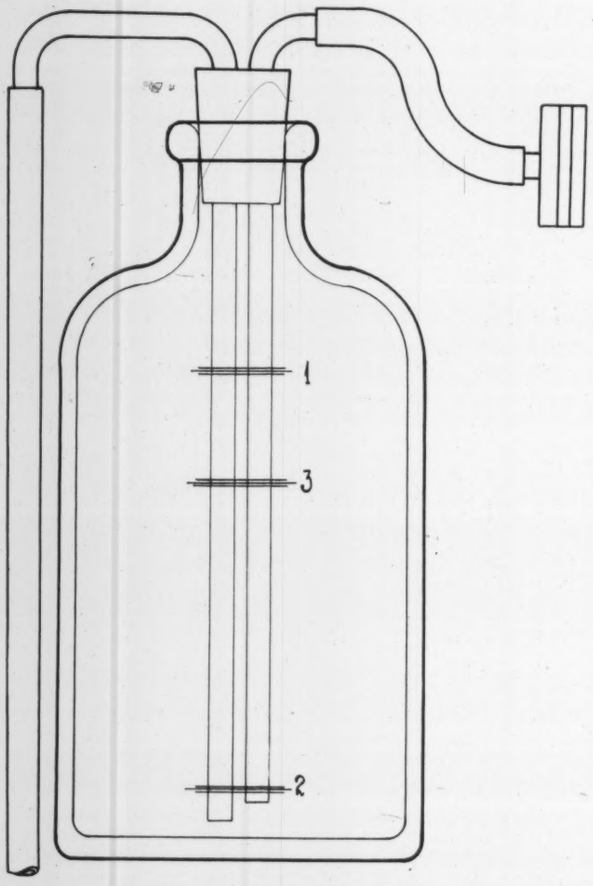


FIG. 2.—MEASURING DEVICE

The variation is so slight that the derivations can be considered verified.

For computation of results from the proposed method, the general equation (4) will be written

$$dV=c\,(A_1-nV)\,dt$$

The length of the siphon being constant, the resistance and consequently also the velocity c through the pores will be constant. Hence

$$ct=\int\frac{dv}{A_1-nV}$$

or

$$ct=\frac{1}{n}ln\left(1-n\frac{V}{A_1}\right)$$

wherein the integration constant vanishes if $t=o$ when $V=o$. From this equation n cannot be solved analytically but it is easy to make up a graph as shown in Fig. 3, from which the solution is obtained.

It is seen that the resistance of the filter paper does not enter into these equations and consequently the length of the siphon may be arbitrarily varied so as to give suitable time readings. However, it was presumed that the clogging is directly proportional to the dust accumulation, which is true only up to a certain accumulation, as evidenced by Miss Ingels' tests. Hence it is important that this accumulation is not exceeded.

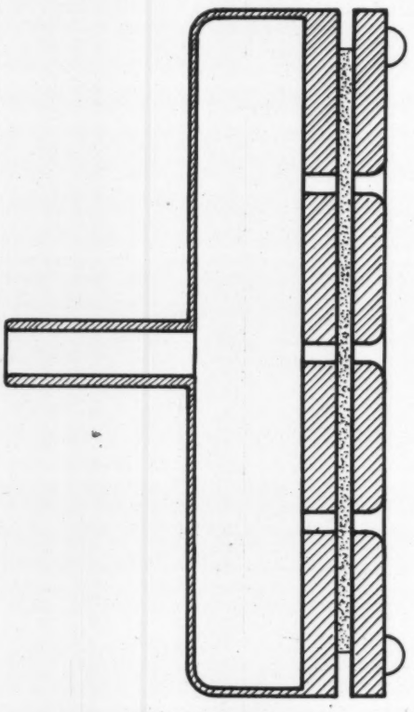


FIG. 1.—FILTER HOLDER

As shown, the dimension of the dust factor n must be L^{-1} in order to satisfy the equations. For simplicity it is assumed to start with that the dust particles consist of cubes of various sizes. The area, obstructed by one of these will be S^2, S being the length of the side of the cube and the total obstruction will be ΣS^2. Accordingly the dust factor n will be

$$n=\frac{\Sigma S^2}{V}\text{ per foot.}$$

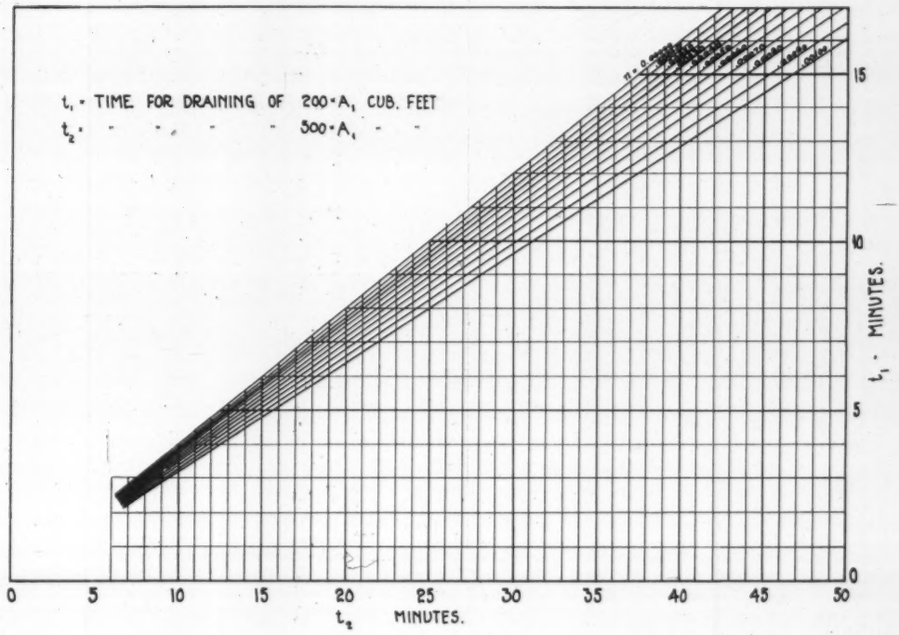


FIG. 3.—GRAPH FOR DETERMINING VALUE OF *n*

The dust factor *n* consequently is a direct measurement of the dust concentration. It expresses the area that would be covered by the dust particles contained in one cubic foot of air.

In reality the shapes of the dust particles are very irregular and there is no physical dimension corresponding to the dust factor *n*, which consequently will have no meaning other than the clogging effect, contained in one cubic foot of air. This effect will generally not express the actual *degree of trouble* caused by the dust, which is the basic weakness of the A-A method of determination.

American Engineering Contributions to World Congress

WHEN the World Engineering Congress opens in Tokio, Japan, in November, over 250 engineers from the United States will attend. The American Committee, of which Elmer A. Sperry is chairman, has made a noteworthy contribution of papers on a variety of engineering subjects.

In the heating and ventilating field, the engineers of the world will learn about the Control of Humidity and Temperature as Applied to Manufacturing Processes and Human Comfort, prepared by a Committee of the Society, headed by Vice-President W. H. Carrier. The Ventilation of Vehicular Tunnels is the subject of a paper by Ole Singstadt, Chief Engineer of the Holland Tunnel, between New York and New Jersey.

A complete list of the papers and titles contributed by American engineers follows:

TITLE OF PAPER	AUTHOR
WELDING AND CUTTING.	Adams, C. A., Director, American Bureau of Welding, Editor-in-chief, care American Welding Society, 33 West 39th St., New York City.
SCIENTIFIC INDUSTRIAL MANAGEMENT.	Alford, L. P., Vice-Pres., The Ronald Press Co., 15 East 26th St., New York City.
HOUSEHOLD REFRIGERATION IN THE UNITED STATES.	American Society of Refrigerating Engineers, 37 West 39th St., New York City.
INSULATION OF HIGH VOLTAGE OVERHEAD TRANSMISSION LINES.	Austin, A. O., Chief Eng., The Ohio Insulator Co., Barberton, Ohio.
FACTORS GOVERNING THE USE AND CONSTRUCTION OF MOTOR TRUCKS.	Bachman, B. B., The Autocar Co., Ardmore, Pa.
STRUCTURAL ENGINEERING AS APPLIED TO MODERN CITY BUILDING CONSTRUCTION.	Balcom, H. G., 10 East 47th St., New York City.
FIFTY YEARS OF PHYSICAL METALLURGY.	Boylston, H. M., Case School of Applied Science, Cleveland, Ohio.
PRECISION MACHINES AND INSTRUMENTS FOR THE MEASUREMENT OF LENGTH.	Burgess, Dr. George K., Director, U. S. Bureau of Standards Washington, D. C.

TITLE OF PAPER	AUTHOR
The Control of Humidity and Temperature as Applied to Manufacturing Processes and Human Comfort.	Carrier, Willis H., Vice-Pres., Am. Soc. Heat. & Vent. Engrs., 29 W. 39th St., New York City.
Handling Materials in Bulk Between Rail and Water.	Case, A. F., The Wellman-Seaver-Morgan Co., 7000 Central Ave., Cleveland, Ohio.
The Development of the Air-Nitrogen Industry in the United States.	Curtis, Prof. Harry A., Prof. of Chemical Engineering, Yale University, New Haven, Conn.
Recent Research on the Thermal Properties of Steam.	Davis, Harvey N., Pres., Stevens Institute of Technology, Hoboken, N. J., and Keenan, Joseph H., care Stevens Institute.
Hydraulic Feeds and Speeds.	De Leeuw, A. L., Consulting Engineer, 1024 Park Ave., Plainfield, N. J.
High Tension Cable Specification and Design in America.	Del Mar, Wm. A., Chief Eng., Habirshaw Cable and Wire Corp., Yonkers, N. Y.
Automatic Train Control.	Ellis, C. E., Secretary, Committee on Automatic Train Control, American Railway Association, 431 S. Dearborn St., Chicago, Ill.
Machinery in the Steel Industry of the United States.	Estep, Frank L., 11 W. 42nd St., New York City.
Resume of the Stability Problem as Applied to Long-Distance Transmission of Power.	Fortescue, Chas. Le G., Consulting Transmission Engineer, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.
Public Health Engineering—A Review of Its Scope.	Fuller, Geo. W., 170 Broadway, New York City, and Eddy, Harrison P., 1300 Stafford Bldg., Boston, Mass., and Phelps, Earle B., College of Physicians and Surgeons, Columbia University, New York City.
Telephone Communication System of the United States.	Gherardi, Bancroft, Vice-Pres., and Jewett, F. B., Vice-Pres., American Tel. & Tel. Co., 195 Broadway, New York City.
1) The Employment of the Less Volatile Fuels for Motor Coach Engines. 2) Heavy Duty Motor Coach Brake Design.	Green, C. A., Vice Pres. in charge of Engineering, General Motors Truck Corp., Pontiac, Mich.
Municipal Engineering in the United States.	Grunsky, C. E., Consulting Engineer, 57 Post St., San Francisco, Calif.
International Co-Operation of Engineers.	Hammond, John Hays, 2221 Kalorama Road, Washington, D. C.
Safety in Dam Construction.	Hazen, Allen, 25 W. 43rd St., New York City.
Application of Motor Transport to the Movement of Freight and Passengers.	Horner, F. C., General Motors Corp., Broadway and 57th St., New York City.
Deflections and Stresses in Pipe Bends.	Hovgaard, Wm., Prof. of Naval Construction, Mass. Inst. of Technology, Cambridge, Mass.
Technical Achievements in Broadcasting and Its Relation to National and International Solidarity.	Institute of Radio Engineers, Alfred Goldsmith, Chairman, 29 West 39th St., New York City.
Economic and Operating Considerations in Railroad Electrification in the United States.	Jackson, Dugald C., Mass. Inst. of Technology, Cambridge. Mass.
Formulation of Rules for the Construction of Steam Boilers and Other Pressure Vessels and for Their Care in Service.	Jacobus, D. S., The Babcock & Wilcox Co., 85 Liberty St., New York City, and Low, F. R., Editor, "POWER," 475 Tenth Ave., New York City.
River and Harbor Engineering in the United States.	Jadwin, Edgar, Lieut.-Gen., ret., former Chief of Engineers, War Department, Washington, D. C.
Light Alloys.	Jeffries, Zay, Aluminum Co. of America, 2210 Harvard Ave., Cleveland, Ohio.
Extra High Voltage Transmission with Special Reference to American Practice.	Jollyman, J. P., Pacific Gas & Elec. Co., 245 Market St., San Francisco, Calif., and Stauffacher, E. R., Electrical Protection Eng., South California Edison Co., Los Angeles, Calif.
The American Skyscraper.	Kahn, Ely Jacques, Park Avenue Building, New York City.
The Business End of Aviation.	Keys, C. M., Pres., Curtiss Aeroplane and Motor Co., Inc., 39 Broadway, New York City.
Modern Blast-Furnace Theory and Practice.	Kinnery, S. P., H. A. Brassert & Co., 310 S. Michigan Ave., Chicago, Ill., and Sweetser, Ralph H., The American Rolling Mill Co., Columbus, Ohio.
The World's Iron Ore Supply.	Leith, Dr. C. K., Chairman, Dept. of Geology and Geography, University of Wisconsin, Madison, Wis.
Ship Vibration.	Lewis, Frank M., Prof. of Engineering, Webb Inst. of Naval Architecture, New York City.
Remote Operating Supervision and Control of Electric Power Stations and Substations.	Lichtenberg, Chester, General Electric Company, 6901 Elmwood Ave., Philadelphia, Pa., and Zogbaum, F., New York Edison Co., 4 Irving Place, New York City.
Some Engineering Aspects of Electric Lighting.	Lieb, John W., The New York Edison Co., 4 Irving Place, New York City.

TITLE OF PAPER	AUTHOR
Chemical Engineering in the United States.	Little, Arthur D., Pres., Arthur D. Little, Inc., Cambridge, Mass.
Irrigation in the United States.	Mead, Elwood, Commissioner of Bureau of Reclamation, Washington, D. C., and Etcheverry, B. A., Prof., 307 Agriculture Hall, Berkeley, Calif.
Cement.	Merriman, Thaddeus, Chief Eng., Board of Water Supply, New York City.
Suspension Bridges with Special Reference to the Philadelphia-Camden Bridge.	Modjeski, Ralph, 121 E. 38th St., New York City.
The Development of the Science of Structural Materials.	Moore, H. F., Research Prof. of Engineering Materials, University of Illinois, Urbana, Ill.
American Developments in Diesel Engines.	Morrison, L. H., Associate Editor, "Power," 10th Ave. at 36th St., New York City, and Katus, Edgar J., Consult. Engr., 203 W. 13th St., New York City.
American Power Station Electrical Equipment.	Newbury, F. D., Mgr., Power Engineering Dept., Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.
Mechanical Problems in the Petroleum Industry.	Nofsinger, C. W., The M. W. Kellogg Co., 225 Broadway, New York City.
The Economics of Water vs. Steam Power.	Orrok, Geo. A., Consult. Engineer, 52 Vanderbilt Ave., New York City.
Resume of Present Methods of Development and Construction of High Voltage Interconnection of Electric Systems in the United States.	Osgood, Farley, 31 Nassau St., New York City, and Mitchell, Wm. E., V. P., Georgia Power Company, Atlanta, Ga., and Downing, P. M., V. P., Pacific Gas & Elec. Co., San Francisco, Calif., and Stone, Edmund C., System Development Mgr., Dusquesne Light Co., Pittsburgh, Pa.
Hydrostatic Uplift Under Masonry Dams.	Parsons, H. De B., 22 William St., New York City.
Lightning.	Peek, F. W., General Electric Co., 100 Woodlawn Ave., Pittsfield, Mass.
The Non-Ferrous Metals from a World Viewpoint.	Probert, Frank H., Dean, College of Mining, University of California, Berkeley, Calif.
Rolled Steel in the Construction of Electrical Machines.	Reist, Henry G., A. C. Engineering Dept., General Electric Co., Schenectady, N. Y.
Petroleum.	Requa, Mark L., 111 Butler St., San Francisco, Calif.
Rapid Transit Subways in New York City.	Ridgway, Robert, 49 Lafayette St., New York City.
Transportation.	Sawyer, Major Mott, Superintendent, Chicago, Milwaukee, St. Paul & Pacific Railway, Port Angeles, Wash.
Marine Engineering and Design.	Seward, H. L., Prof. of Mechanical Engrng., Yale University, New Haven, Conn.
Ventilation of Vehicular Tunnels.	Singstad, Ole, Chief Eng. and Superintendent, The Holland Tunnel, New York City.
Standardization.	Skinner, C. E., Asst. Director of Engineering, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.
Quantity-Rate Fluid Meters.	Smith, Ed. S., Jr., Builders Iron Foundry, 9 Codding St., Providence, R. I.
Development in the American Coal Industry, 1913-1928.	Smith, Dr. Geo. Otis, Director, U. S. Geological Survey, Washington, D. C., and Tryon, F. C., U. S. Bureau of Mines, Washington, D. C.
Recognition of the Engineer and the American Engineering Societies.	Sperry, Elmer A., The Sperry Gyroscope Co., Manhattan Bridge Plaza, Brooklyn, N. Y.
Railway Engineering.	Stevens, John F., 30 Church St., New York City.
Propeller Design Developments.	Taylor, D. W., Rear Admiral (C. C.) U. S. N., ret., 1811 Nineteenth St., N. W., Washington, D. C.
Refrigeration in the Preservation of Food.	Taylor, H. F., V. P. for Scientific Research, Atlantic Coast Fisheries Co., 16 Exchange Place, New York City.
Settlement Analysis—the Backbone of Foundation Research.	Terzaghi, Dr. Chas., Mass. Inst. of Technology, Cambridge, Mass.
Operation of Turbo Generators on a Total Temperature Rather Than a Temperature Rise Basis.	Torchio, Philip, New York Edison Co., 130 E. 15th St., New York City, and Foster, W. J., General Electric Co., Schenectady, N. Y., and Newbury, F. D., Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.
Petroleum Production Engineering.	Umpleby, J. B., Pres., Goldelline Oil Corp., Braniff Bldg., Oklahoma City, Okla.
Development and Trends in Standardized Quality Production.	Van Deventer, John H., 11 West 42nd St., New York City.

TITLE OF PAPER	AUTHOR
Engineering as Applied to Agriculture.	Walker, Prof. H. B., Div. of Agricultural Engineering, University of California, University Farm, Davis, Calif.
Needs and Problems of Aeronautic Engineering.	Warner, E. P., Editor "Aviation," 52 Vanderbilt Ave., New York City.
Industrial Progress Made Through Research and Its Economic Importance.	Whitney, Willis R., Research Laboratory, General Electric Co., Schenectady, N. Y.
Engineering Education in America.	Wickenden, Wm. E., Society for the Promotion of Engineering Education, 29 West 39th St., New York City.
Design of American Passenger Automobiles.	Wolf, Austin M., 316 Ferry St., Newark, N. J.
Improved Utilization of Railroad Equipment.	Wright, Roy V., Managing Editor, "Railway Age," 30 Church St., New York City.

A. S. M. E. Fuels Meeting in Philadelphia

On October 7 to 10, the Third National Fuels Meeting of the *American Society of Mechanical Engineers* will be held in Philadelphia, Pa., and engineers interested in combustion problems are invited to attend the three days sessions to be held at the Bellevue-Stratford Hotel.

The program is divided into five sections covering general topics of fuel uses and conservation, power plant practice, industrial and domestic applications and smoke abatement. In the symposium on domestic heating, the following subjects will be covered:

Economics of House Heating and Insulation, H. B. Johns, Peoples Gas Light & Coke Co., Chicago, Ill.
Gas, M. J. Roberts, Pres., Roberts Gordon Appliance Co., Buffalo, N. Y.
Oil, H. F. Tapp, Technologist and Asst. Secy., American Oil Burner Assn., New York City.
Electricity, A. R. Stevenson, Jr., Engr. Gen'l Dept., General Electric Co., Schenectady, N. Y.
Anthracite, A. F. Duemler, Mgr. Anthracite Coal Service, New York City.
Bituminous, F. R. Wadleight, Cons. Engr., New York City.
Coke, P. Nicholls, U. S. Bureau of Mines, Pittsburgh, Pa.
Some Fundamentals of the Domestic Heating Problem, Samuel S. Wyer, Cons. Engr., Columbus, Ohio.
Stokers for Apartment Houses and Office Buildings, Joseph Harrington, Pres. Joseph Harrington Co., Riverside, Ill.

Among the contributors to the smoke abatement session is Philip Drinker, Asst. Prof. of Ventilation and Illumination, Harvard University, who will speak on the Effect of Pulverized Fuel Ash on Penetration of Ultra Violet Rays.

The committee in charge of this session is headed by Victor J. Azbe, Chairman, Executive Committee, Fuels Division, A. S. M. E., and H. A. Hoffer, Chairman Philadelphia General Arrangements Committee.

Eighth National Power Show

The eighth National Exposition of Power and Mechanical Engineering will open at Grand Central Palace, New York City, December 2, and continue throughout the week.

The National Power Show, as the exposition has come to be more briefly known, has become a national institution since its inception in 1922, when the attendance was only 47,589. Last year's attendance had risen to 123,000 and the number of exhibitors had made the phenomenal showing of 542 in 1928 as against only 105 in the first year of the exposition.

This growth, it is pointed out, is due largely to the fact that the exposition covers the entire field of power generation and mechanical engineering. It represents a nation-wide gathering of exhibitors, carrying a complete collection of machinery and products intimately related to the subject.

Another important factor entering into the remarkable growth of the National Power Show can be traced to new developments in the field. Never in the history of science has there been such rapid and exceptional advancement in mechanical engineering equipment as during the years the exposition has been functioning.

There have been more inventions and more improvements in these few years than ever before, and it is these improvements and inventions which will furnish stimulus and zest in the competitive reciprocity between producer and consumer at the exposition.

From a list of over 400 corporations which have already reserved space, this year's exhibits may well be grouped into three general classes: first, new developments, comprising new inventions, new processes and new ideas; second, improvements on old principles, and third, those grouped under engineering staples.

However, progress, as represented by the first group is the inherent characteristic of what may be expected at this year's show. New developments and better ways of doing things, side by side with the staples, produces a sort of melting pot where visitors and exhibitors are afforded an opportunity of becoming better acquainted.

Producer and consumer meet under unique conditions to see, to compare and to understand the different products and processes. The National Power Show disseminates knowledge of mechanical advancement for the benefit and inspiration of those both inside and outside of the engineering profession. It keeps one posted on the latest developments and methods of operation and affords an opportunity of actually seeing the equipment and instruments in action.

Coincident with the National Power Show, the *American Society of Mechanical Engineers* will hold their annual meeting in New York City.

The Advisory Committee of the National Power Show is made up of the following well known engineers: I. E. Moultrop, *chairman,* Edison Electric Illuminating Co., Boston; Homer Addams, past president, American Society of Heating and Ventilating Engineers; A. Black, chairman, professional divisions, A. S. M. E.; N. A. Carle, general manager Pacific Electric & Manufacturing Co.; Fred Felderman, past national president, *National Association of Stationary Engineers;* F. M. Gibson, chairman, power division, *A. S. M. E.*; C. F. Hirshfeld, chief of research department, Detroit Edison Co.; O. P. Hood, chief mechanical engineer, United States Bureau of Mines; John H. Lawrence, Thomas E. Murray, Inc.; Thornton Lewis, president, American Society of Heating and Ventilating Engineers; Fred R. Low, past president, *American Society of Mechanical Engineers*; David Moffat Myers, consulting engineer; M. S. Sloan, president, *National Electric Light Association*; Arthur J. Wood, president *American Society of Refrigerating Engineers.*

Fred W. Payne and Charles F. Roth will manage the National Power Show.

Stoker Equipment and Furnaces

Stoker Equipment and Furnaces is the subject of publication No. 289-71, serial report of the prime movers committee for June, 1929, engineering national section of the *National Electric Light Association,* 420 Lexington Ave., New York, N. Y. The report is well illustrated with photographs and line drawings and diagrams.

School Room Ventilation[1]

By A. C. Willard[2], Urbana, Ill.

Introduction

THE entire field of ventilation presents no more important problem than the adequate and satisfactory ventilation of the average school room, occupied for approximately six hours of the day by a class of 40 pupils and a teacher. In such an atmospheric environment as may exist in these rooms during the winter months, most of us have acquired our formal education, and some of us have continued on as educators. In so far as the atmospheric environment of the classroom affects the mental activity or development and the physical health of the occupants, the problem is of vital importance and universal interest. Judging from the many discussions of this matter and the numerous investigations, past, present and projected, a final and entirely satisfactory solution has not yet been obtained. Possibly such a solution is not attainable within practical limits of cost, when we consider the extreme variability of individual temperaments and sensations.

Objectives

It is a fairly simple matter to set up the objectives sought in any adequate system of school room ventilation. They are, briefly, the comfort and health of the occupants. These objectives should be accomplished automatically and unobtrusively, leaving both pupils and teacher free to pursue the educational program in hand, without distraction from bodily sensations of heat and cold or disturbing drafts. Most certainly the overworked teacher should be relieved of the duties of a janitor, or the mechanical functions of a thermostat.

Methods

The present *state of the art* provides every conceivable type of equipment, ranging from a single stove in a one-room school to the most elaborate mechanical systems of heating and ventilation in schools of fifty rooms or more, including high pressure boilers, engines, motors, fans, ducts, indirect and direct radiators, air washers, air filters, ozone generators and all the automatic regulating devices known to the engineer. Rather recently the mechanical systems have been decentralized to some degree in certain installations, and *small* indirect fan units have been spread over the buildings, eliminating most of the air supply ducts and indirect heaters of the central systems. Still more recently, a *back to nature* movement has developed, advocating *open windows* with no air supply ducts to the classrooms, which are merely equipped with direct radiation placed under windows fitted with window boards, and gravity vent ducts are relied on to assist in providing for a more or less indefinite air change.

The school room ventilation problem has become very complicated and the *best* solution for any given case is not only giving school boards, public health officials, architects and engineers a great deal of concern, but also there are many widely different opinions as to what determines which is the *best* solution. Beyond the shadow of a doubt, there is no *best,* as conditions always alter cases, but certain basic factors must be reckoned with in making comparisons between and selections of these systems.

The Essential Factors in School Room Ventilation

Thanks to the basic work of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, in conjunction with the U. S. Public Health Service and the U. S. Bureau of Mines, during the past five years, we have definite information on the essential factors affecting human comfort in any given atmospheric environment. These data, together with the studies of the *American Public Health Association* on the incidence of various respiratory diseases among school room occupants, serve as valuable criteria for proving or disproving the claims of any *best* system of school room ventilation.

We know today that three basic factors determine human comfort, in so far as our atmospheric surroundings are concerned. These are: (1) air temperature, (2) relative humidity and (3) air movement. No one of them is independent of the other two. That system which will *automatically control* these three conditions, which must be kept in the proper relation to each other, and at the same time promote the health of the occupants, is the *best* system of school room ventilation. No one familiar with the problem is likely to question the preceding statement. But the success and methods with which the various systems provide these conditions and achieve the objectives of comfort and health are a matter of great argument, most of which hinges on the source and amount of air introduced and the method in which it is supplied to the school rooms.

Sources, Amount and Method of Supplying Air

In the simpler systems using stoves or direct radiators and open window, where no indirect radiation is provided, the air is brought in *raw* and untreated, directly from out of doors, and mixed with the air of the school room. In the mechanical systems, the *raw* out-of-door air is first tempered by indirect radiators, and then cleaned, humidified and reheated to the desired room inlet temperature, all of which is accomplished automatically. Frequently some air is recirculated from the building, especially in very cold weather, and mixed with the outside air before it enters the cleaning and conditioning apparatus.

The amount of air is extremely variable in the first case, depending on both the weather or wind direction and intensity, and the interest and attention of the teacher to this matter, but in the second case it is under

[1] Presented before the 57th Annual Meeting of the *American Public Health Association*, Chicago, Ill.

[2] Professor of Heating and Ventilation and Head of the Department of Mechanical Engineering, University of Illinois, Urbana, Ill.

positive mechanical control at all times, so that the teacher has no function to perform but to teach. In the former case, little or no outside air will be taken in during very cold weather, and the proper relation between temperature, humidity and air motion may be very bad. On mild days, however, conditions may be quite satisfactory. In the latter case, with a fixed amount of air constantly supplied, the room air conditions are uniform in all rooms and independent of the weather and the teacher. The determination of the correct amount of air, therefore, becomes a most vital and important matter in the latter case, since pupils and teacher are unable to control or vary the amount of air introduced.

With the *open window* systems the method of supplying air is simple and direct, and has no effect on its out-of-door qualities, whatever they may be. With the mechanical system the method of supplying air is more complicated and several changes are wrought on the out-of-door qualities of the air before it finally enters the school room. There may be some vital property (ionization has been suggested but not yet identified as the essential one) in out-of-door air which is altered by the heaters and ducts of certain types of mechanical systems of ventilation, especially those operating without suitable air conditioning apparatus. It is more likely that this altered quality is merely a manifestation of the result of overheating in those school rooms supplied with an unnecessarily large quantity of air which is too dry. See item (2) below.

The Results

Measured in terms of the effect produced in the school room as reflected by the comfort and health of the occupants, the following conclusions may be set down.

1. Studies based on respiratory illness, as shown by absenteeism and infected pupils in attendance, have shown generally better results for an intelligently operated *open window* system than for *some* types of mechanical systems operating without proper humidification when supplying the generally accepted 30 cu. ft. of air per pupil per minute.

2. The introduction of large quantities of dry air by mechanical means, especially where there are no air washers, necessitates higher air temperatures in the room to overcome the sensation of coolness produced by draftiness and rapid evaporation from the bodies of the occupants.

3. While comfort and health studies in school rooms have shown that incomplete or inadequate mechanical systems based on the antiquated CO_2 theory of supplying 30 cu. ft. of all outside air per minute per pupil have been found to have certain shortcomings, it does not follow that a mechanical system, as such, is unsuited to proper and effective school room ventilation. It would appear rather that a mechanical system designed on the basis of our present knowledge of the really essential factors affecting ventilation is, in most cases, the *best* present solution of this problem, if complete and positive automatic control of the proper air conditions in a school room is to be accomplished.

4. The ratio of room volume per pupil appears to have an important effect on the results secured with the *open window* system. High values of this ratio are desirable, if not absolutely essential in this case. Incidentally, a much wider distribution of the direct radiator surface becomes imperative, and much larger gravity vent flues are essential.

5. It would also appear that any attempt to place the responsibility for the entire control and regulation of the heating and ventilating system on the teacher is of dubious value, when we stop to consider that, with constantly changing outside temperatures and winds, no possible adjustment of window openings can be correct for any length of time even for a given room. Considering the building as a whole, it will be apparent that rooms on the windward side are always subjected to pressure, while those on the leeward side are subjected to suction whenever an appreciable wind is blowing. Outside air will readily enter the open windows of the former rooms, but inside air will pass out of the open windows of the latter rooms, so that *open window* ventilation under windy weather conditions can be applied to only one-half of the building at best. The leeward half has to take the "drift" of the air from the rooms of the other half.

6. A proper comparison of economic results is not easily made, although installation costs of the actual heating and ventilating equipment are obviously less for the *open window* system than for the complete mechanical system, even though the latter supplies a much more conservative quantity of air than in the past. In operation, for the same amount of outside air supplied the classrooms, there should be no essential difference in fuel requirements. Motor operation will, of course, add to the operating cost of the mechanical system in which each room is always provided with a positive air supply, which is the prime requisite of controlled ventilation.

Conclusions

As was stated at the outset, possibly no entirely satisfactory solution of the school room ventilation problem is possible by any system within the practical limits of cost. Extreme simplicity without positive air supply and automatic regulation places an unwarranted responsibility on the teacher who must constantly adjust the windows to the vagaries of the weather.

Complete mechanical systems will automatically maintain any desired atmospheric condition found essential to comfort and health in the classroom. Failures in the past, resulting from incomplete knowledge of the essential factors only recently identified, are no satisfactory index of comparison between what has been and what will be.

The further discussion and investigation of this important matter should be pursued even more actively than in the past, to the end that the *best* system may prevail.

Hauss Goes to Holland

Announcement is made that Charles F. Hauss, formerly special representative for the Far East of the National Radiator Co., Hull, Eng., a subsidiary of the American Radiator Co., New York, has become identified with the N. V. Radiatoren, Amsterdam, Holland, also a subsidiary of American Radiator Co.

J. P. Cosgrove, New York, of the Export Department of the American Radiator Co., will take over the duties of Mr. Hauss temporarily until his successor in the Far East is appointed.

PRESIDENT'S PAGE

DURING the summer months our Society and its activities have been in my thoughts a great deal. One idea which stands out so preeminently above all others is the loyalty which our members have to the Society and its work.

When I think of the immense amount of committee work done by our members, without any pay and often with expense to themselves, I glow with pride at the unselfishness of my fellow engineers.

Indeed, such unselfish devotion has accomplished much — my mind dwells on THE GUIDE, our papers in THE JOURNAL, the Code of Minimum Requirements, various codes for testing apparatus, the management of the Society affairs through the standing committees, and last but not least, the cooperation with the Research Laboratory through the many advisory committees.

While the above thoughts may have come to many of our members, I wonder if they have thought of the loyalty of numbers of our members in foreign lands.

This summer, through the courtesy of one of our members in Paris — Mr. A. Beaurrienne — I had the pleasure of meeting and dining with the President of the *Association des Ingenieurs de Chauffage et Ventilatun de France* (French Institute of Heating and Ventilating Engineers), and the President of the *Heating and Piping Contractors Association of France*.

Now, Mr. Beaurrienne's loyalty doesn't stop with paying graceful courtesy to our officers, thoughtful as that may be, but he translates our papers and proceedings into French and the more important ones are translated by him and delivered as a part of his lectures in the Technical School at the University of Paris.

Mr. Beaurrienne is not a professor nor is he even engaged in educational or editorial work as one might think, but he is a hard headed business man engaged in manufacturing, selling and contracting work.

Again, in England, I found members who had to their credit thirty odd years of allegiance to our Society. It was my great pleasure to be present as honor guest at a dinner sponsored by Mr. W. Nelson Haden, and given by the English members of our Society and the officers of the *Institution of Heating and Ventilating Engineers*.

It was a most enjoyable affair, including many of the formal customs of old England, such as toasting with real old wines, the King, the President of the United States, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and others present.

Mr. Haden, whom many of our members will recall met at our Washington meeting in 1923, is not only a member of long standing in our Society, but is a past President of the *Institution of Heating and Ventilating Engineers of Creat Britain*.

At this dinner I had the pleasure of presenting the resolutions adopted by the Society at our last summer meeting at Bigwin Inn, felicitating our fellow engineers in England, which resolutions our efficient secretary had had engrossed most beautifully with illuminated letters on craft paper.

I found our fellow engineers in foreign lands everywhere, but especially in England, vitally interested in our Research work and in THE GUIDE. Their remarks were most appreciative as well as complimentary regarding the work we were doing along these lines, even though their every day problems in heating and ventilating are quite different from ours, made so largely by the difference in climatic conditions.

Mr. J. Roger Preston, President of the *Institution of Heating and Ventilating Engineers*, and several fellow members were kind enough to arrange an inspection trip, on which they accompanied me, to the Building Research Station at Garston, near Watford, England, where considerable work is being done by the Department of Scientific and Industrial Research of His Majesty's Government, on the heating and ventilating portion of which a committee of the British Institution of Heating and Ventilating Engineers is cooperating in an advistory capacity.

Here, I had the pleasure of meeting Mr. A. F. Dufton, M.A., D.I.C., and Mr. P. W. Barnett, who are in immediate charge of the work on heating and ventilating, and who were kind enough to conduct our party over the buildings and describe in detail their methods and results.

This particular work has been under way more than a year and it includes a minute study and comparison of the effect of heating solely by radiation and heating by the combination of radiation and convection.

The originality and resourcefulness of the staff shows itself in the many novel instruments, most of which are electrical, developed for securing and recording the various data.

It will be well for our members to acquaint themselves with this work, for I feel sure some novel and vitally important results will be secured. I have arranged for us to receive at our headquarters in New York, a copy of all of their publications and I hope it will be possible for us to reprint in our JOURNAL their most important conclusions.

Thus, other engineers, as well as our own members, are busily engaged in splendid efforts striving to utilize the laws of nature and increase the sum of human knowledge.

It is only possible at this time for me to touch briefly on these various items which show so clearly the devotion and loyalty of our various members everywhere to the ideals for which our Society was founded.

I hope, however, it may be sufficient to have us all feel a stronger faith in the worthwhileness of our undertaking and a renewed resolve to continue to do our utmost to uncover still more knowledge for the benefit of mankind.

New York Commission of Ventilation Suspends

Information has been received that the New York Commission of Ventilation will discontinue its work September 30, 1929 and close its headquarters at 370 Seventh Ave., New York City.

The Commission, as at present constituted, became active in 1926 and was the unofficial successor to the New York State Commission of Ventilation appointed in 1913 by Governor William Sulzer of New York. This New York State Commission of Ventilation was composed of C. E. A. Winslow, Edward L. Thorndike, James A. Miller, Frederic S. Lee, Earle B. Phelps and D. D. Kimball. This Commission continued its investigations until 1917 and had its report published in 1923.

The same members who composed the original Commission with the exception of Dr. James A. Miller, reorganized in 1926 under the name of the New York Commission of Ventilation. Dr. Rufus Cole of Rockefeller Institute replaced Dr. Miller and the new group was enlarged by the appointment of Dr. George T. Palmer and an executive secretary, Thomas J. Duffield.

The work of the original Commission was done in the laboratory of the College of the City of New York and observations were made in several schools of the city. This work was summarized in the Commission's report and was commented on by a special Society committee at the semi-annual meeting 1923 (TRANSACTIONS, Vol. 29, 1923, p. 215).

The work of the unofficial group consisted of a series of field tests in schools at Syracuse, N. Y., New York City and elsewhere and the most recent program of investigations is otherwise in the JOURNAL of the Society. (Vol. 34, September 1929, p. 735).

British Engineers Hold Summer Meeting

The summer meeting of the *Institution of Heating and Ventilating Engineers* was held at the Spa Hotel, Bath, England, on June 17, 18 and 19, 1929. The first day was devoted to a Council Meeting; the second morning to technical subjects and the last day to sightseeing and recreation.

The program included a paper entitled Heating Research at Watford, 1928-1929, by Dr. Margaret Fishenden and A. F. Dufton of the Building Research Station, Department of Scientific and Industrial Research.

The Bath Oliver Biscuit Factory was visited by the members and guests attending the convention, as well as the Mineral and Roman Bath and the Abbey.

The entertainment program included a motor drive via Wells, Priddy and the Cheddar Gorge, to Cheddar.

The banquet was held in the main dining room of the Hotel, June 18, 1929, after which dancing and a concert in the Ball Room took place.

British Engineers Entertain President Lewis

A complimentary dinner to Pres. Thornton Lewis was given by the *Institution of Heating and Ventilating Engineers* on July 18, 1929, at the Connaught Rooms, Great Queen Street, London.

The dinner was arranged by W. Nelson Haden, a member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and also a member and past-president of the *Institution of Heating and Ventilating Engineers.*

This dinner was attended by the British members of our Society, and J. Roger Preston, President of the *Institution of Heating and Ventilating Engineers,* as well as the officers and council of that organization.

A splendid reception was given President Lewis on this occasion, and the hospitality of our colleagues across the sea was greatly appreciated by him.

International Heating and Ventilating Exposition, Philadelphia, January 27-31, 1930

An invitation to the *Institution of Heating and Ventilating Engineers of Great Britain* has been issued by H. P. Gant, chairman of the Advisory Committee for the First International Exposition of Heating and Ventilating, was forwarded to Pres. Thornton Lewis, of the A. S. H. V. E. after he had left for England so that he could deliver the invitation in person. It is hoped that as many as possible of the British engineers will visit the exposition, and attend the Annual Meeting of the Society in Philadelphia, Pa., January 27-30, 1930.

An invitation will also be sent to 66 foreign government officials, which it is hoped will also increase the number of foreign visitors to the exposition to be held in Commercial Museum, Philadelphia, January 27-31.

It is planned to have a number of educational exhibits, showing various stages of the art of heating and ventilating. The Committee is desirous of displaying exhibits of the inventive genius of Benjamin Franklin with respect to the heating of buildings, and any information concerning the availability of heating equipment designed by Franklin or other pioneers in this field, of historic interest, will be greatly appreciated.

Interest in the heating and ventilating exposition is manifest and a large portion of the available space has been reserved by manufacturers. Already 144 have signified their intention of installing exhibits which will be of interest to heating and ventilating engineers, architects, contractors, building and home owners and people engaged in the construction industry. Plans for a large attendance at the Commercial Museum are being made for it is anticipated that like other important industrial expositions a large public attendance will be in evidence.

The Commercial Museum where the exposition is to be held adjoins the grounds of the University of Pennsylvania and is easily accessible from Philadelphia railroad stations.

The Commercial Museum houses all of the large expositions held in Philadelphia and is admirably fitted for the purpose as all of the exhibits will be on one floor and will make an impressive showing.

An advisory committee of the Society is working with the exposition management and is composed of the following members: H. P. Gant, *chairman,* E. B. Langenberg, J. I. Lyle, J. F. McIntire, F. R. Still, H. L. Whitelaw, E. K. Webster, H. C. Murphy, A. S. Armagnac, D. S. Boyden, A. C. Edgar, C. V. Haynes, W. H. Carrier and Roswell Farnham.

Philadelphia Chapter

The Philadelphia Chapter of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has just gone through one of its most successful years, having made a substantial increase in the membership of the Society and improved its financial position, acording to the report of A. J. Nesbitt, Secretary. At the close of 1928 there were 174 regular members and 65 chapter members. Every effort is being made to induce as many of the Limited Chapter Members as possible into the regular membership of the Society.

At the February 1928 meeting T. E. Crone spoke on the *Development and Use of Brass Radiation.* The subject was very interesting and the meeting was well attended.

The March 1928 meeting was one of the most successful meetings in the history of the local chapter, at which meeting Prof. A. C. Willard of the University of Illinois and then President of the American Society of Heating and Ventilating Engineers, spoke on the *Ventilation Investigation of the Holland Tunnel.*

D. Knickerbacker Boyd, Architect, had charge of the April 1928 meeting and with the assistance of W. R. Falck, T. J.

Quinlan, E. C. Lloyd and Dr. P. A. Houseman, gave a very interesting presentation on the subject of *Insulation of Homes and Methods of Application.* This meeting was very well attended.

At the May 1928 meeting S. R. Lewis, then Chairman of the Committee on Research, made a very fine impression on the local chapter with his interesting talk on *Commercializing Professional Knowledge.*

In accordance with the usual practice there was no meeting in June, July, August or September, 1928.

The speaker of the October 1928 meeting was E. B. Langenberg of St. Louis, whose subject was *What Does a Heating Man Sell.* Mr. Langenberg illustrated his speech with some of his own experiences in the engineering field and provided a very interesting meeting for the chapter.

At the November meeting, F. C. Houghten spoke on the activities of the Research Laboratory of the Society. Mr. Houghten illustrated his talk with a number of interesting slides, and the meeting was well attended.

Prof. J. E. Emswiler of the University of Michigan presented an interesting talk at the December 1928 meeting on *Some Theoretical and Practical Factors in Universal Ventilation.* At the same meeting W. C. Randall, Chief Engineer of the Detroit Steel Products Co. spoke on *The Aeration of Typical Industrial Buildings.* Both speakers illustrated their talks with slides.

At this meeting J. D. Cassell, often called the Father of the Philadelphia Chapter, was appointed as Honorary Chairman of the 1930 Annual Meeting Committee.

F. D. Mensing was nominated for member of the Nominating Committee of the Society, and R. C. Bolsinger as alternate. The nomination was properly seconded and carried.

The following are the officers and Board of Governors for the year 1929:

OFFICERS:

President—A. C. Edgar
Vice President—H. G. Black
Secretary—L. C. Davidson
Treasurer—E. N. Sanbern

BOARD OF GOVERNORS:

R. V. Frost A. J. Nesbitt Lee Nusbaum

Minnesota Chapter

August 9, 1929. The new officers for the Minnesota Chapter for the 1929-30 season are announced and are as follows: *President*—Edwin F. Jones, 301 Zenith Bldg., St. Paul, Minn.; *Vice-President*—D. M. Forfar, 240 Seventh Ave., S., Minneapolis, Minn.; *Secretary-Treasurer*—M. S. Wunderlich, Flaxlinum Insulating Co., Hampden & Wabash Ave., St. Paul, Minn.

A successful year is anticipated, and the enthusiasm over the activities planned is unanimous among the members of this chapter.

Death of Ralph A. Fleming

Word has just reached us of the death of Ralph A. Fleming, of Columbus, O., on May 16, after an illness of more than a year. Mr. Fleming was secretary and heating engineer of Floyd L. Benedict, Inc., and during his business career, which began in 1906, he was with the Columbus Brass Co., Rose & Cox; Standard Sanitary Manufacturing Co., and Pierce, Butler and Pierce Mfg. Corp.

A number of the business and industrial buildings in Columbus have heating and ventilating plants designed by Mr. Fleming, who spent most of his business career there.

He was born February 20, 1885, at Grand Rapids, Mich., and received his early education in that city.

Mr. Fleming is survived by his widow and five small children to whom the officers and Council extend their sincere sympathy.

Death of Luther B. McMillan

As a result of injuries suffered in an airplane crash near the Newark, N. J. Airport on August 9, Luther B. McMillan died the following day at the St. James Hospital, Newark.

Since 1916, Mr. McMillan has been general consulting engineer for the Johns-Manville Corp., New York City, where he handled all the engineering problems connected with heating and power plant work. For a good many years he had specialized in the field of heat transmission with particular attention to heat transfer through insulation and was well known to Society members for the numerous contributions of papers that he made on these subjects.

LUTHER B. MCMILLAN

Mr. McMillan was born September 21, 1891 at Bem, Mo., and attended the Texas Agricultural and Mechanical College, graduating in 1911 with a B. S. degree. In 1912 and 1913 he was an instructor in mechanical engineering and took graduate work receiving the degree of M. E. in 1912 and Chem. E. in 1913. He was awarded a fellowship in engineering at the University of Wisconsin and in 1914 received the degree of M. S. from that University. After serving two years as instructor in steam and gas engineering, specializing in heat transmission research, he joined the Johns-Manville organization.

During his 16 years' service with the Johns-Manville Corp. he made a splendid record in the insulation field and the testing laboratory of the corporation at its Manville factory, regarded as one of the finest private laboratories in the U. S., was established under Mr. McMillan's direction.

Mr. McMillan was a member of the A.S.H.&V.E.; A.S.M.E.; A.S.R.E.; The Engineers' Club of New York; Western Universities Club; and the Texas Agricultural & Mechanical College Club of New York.

At the time of his death Mr. McMillan was a member of the Council of the A.S.M.E. and served on several of its important committees.

His membership in the A.S.H.&V.E. dates from 1918 and during the past 10 years he has contributed numerous papers which appear in the TRANSACTIONS. He has prepared the Insulation chapter for the A.S.H.&V.E. GUIDE on several occasions. He served on a number of the Society's technical committees and was prominently identified with the work of the New York Chapter.

Mr. McMillan's writings were prolific and he prepared numerous papers for other engineering societies including the *A.S.M.E.;* Iron & Steel Electrical Engineers Society; and various handbooks such as Marks, Kent and that of the *National District Heating Association.*

Because of his outstanding work he served with the National Research Council and from 1924 to 1928 was on the Sub-

Committee of Heat Transfer. Upon the organization of the Committee on Heat Transmission he was elected a member of the executive committee and served as chairman of the sub-committee on Thermal Insulation.

Mr. McMillan was an aviation enthusiast and had great confidence in the future of that industry. He was a licensed pilot and frequently utilized this method of transportation in travelling over the wide range of territory covered by his professional activities.

Mr. McMillan is survived by his widow, Helen E. McMillan and two young children as well as by his father, sister and brother.

In his death the engineering profession has lost one of its brilliant representatives and the Officers and Council join with the members in expressing their profound sorrow to his family in their recent bereavement.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the name of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 9 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned. All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by October 1, 1929, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary, immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
Brooks, Samuel J., Chief Engr., Spencer Heater Co., Toronto, Ont., Canada.	Grant E. Cole J. O'Neill	Harry H. Angus Arthur S. Leitsch
Crandall, R. Kay, Buick Motor Co., Flint, Mich.	M. S. Titus	Jas. Watt W. K. Ash
Hooper, Vernon Fowler, Htg. Engr., Lord & Burnham Co., Irvington, N. J.	Arthur P. Weiss A. E. Bastedo	Harry W. Fiedler
Park, Clifton D., Dist. Sales Repr., Carrier Engrg. Corp., Boston, Mass.	Herman Worsham L. Logan Lewis	Harry L. Janet A. E. Stacey, Jr.
Peterson, Adolph J., Service & Combustion Engr., Anthracite Coal Service, Chicago, Ill.	Homer R. Linn H. G. Kreissl	Carlyle M. Terry
Pocock, Paul C., Mgr., New England Branch, York Heating and Ventilating Corp., Boston, Mass.	H. P. Gant C. D. Graham	Jas. V. Cavileer Donald French
Randolph, Charles H., Kewanee Boiler Corp., Milwaukee, Wis.	Harris R. Cook R. G. Olson	H. M. Miller W. H. Wilson
Roberts, Edward F., Jr., Engrg. Draftsman, Haynes Selling Co., Inc., Philadelphia, Pa.	Merrill O. Blaikin Raymond E. Jones	Franklin W. Wandless A. E. Kriebel
Heim, E. Frank, Consulting Engr., Architects Bldg., 1800 E. St., N. W., Washington, D. C.	F. R. Still Herbert E. Barth	Benj. Adams Raymond E. Jones

Candidates Elected September, 1929

In the past issues of the Journal of the Society there were posted among the list of Candidates for Membership 7 candidates for election in the different grades of Membership, whose names have been balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

ASSOCIATES

Darling, Arthur Balfour, Darling Bros., Ltd., Montreal, Canada.

Griffiths, Morgan Reginald, Mgr., Toronto Branch, Canadian Blower & Forge Co., Ltd., 146 King St., W., Toronto, Ont., Canada. (Reinstatement.)

Hillen, William George, Carrier Engineering Corp., Newark, N. J.

Strickland, Albert W., Owner & Mgr., A. W. Strickland Plumbing, Heating & Electrical Co., Big Timber, Mont.

JUNIORS

Grant, Walter Adams, Carrier Engineering Corp., Newark, N. J.

Hillen, Alvin G., Carrier Engineering Corp., Newark, N. J.

True, John Edward, District Mgr., York Heating & Ventilating Corp., 622 Broadway, Cincinnati, O.

Heating · Piping and Air Conditioning

EDITORIALS

Technical Interests Stimulate Business

There are three well defined groups among our readers and it is a happy coincidence that a technical journal such as this can not only serve these groups in an informative manner but in a business building way as well. This is because an article that gives one group an idea for modernizing or changing an existing system of heating, piping or air conditioning, or for meeting a problem that needs solution, will lead to work that might mean a contract for one of the other groups.

To explain—our readers consist of consulting engineers, of heating, piping and air conditioning contractors and of engineers responsible for the maintenance and operation of this type of equipment in industrial plants, office buildings, hotels, institutions, schools, etc. One of our contributing editors writes an article on, say, heating an industrial plant. In this article the method, application and results of a system of industrial heating are covered in an engineering and informative way. It is read by our engineer subscribers in industrial plants. They might be having trouble with their heating and the article suggests a solution. Or, they might see in the article an opportunity for savings not being effected in their present system, and such savings might justify changes that will make them possible.

Such changes might be handled by the plant organization itself. On the other hand, if it is a major undertaking, chances are that some outside contractor will land the job, and it might also mean that some consulting engineer will be retained to design the system. Thus, a technical article, aimed to give information, not only gives the information but in addition suggests an idea that leads to a job. This is profitable for the consulting engineer, for the contractor and, of course, for the operating engineer, as it is his responsibility to keep his plant running uninterruptedly and efficiently.

Another article might describe an air conditioning installation. At the time it is presented it is conceivable that a client of a consulting engineer or contractor has a manufacturing process problem or a problem concerning the health and comfort of his employees. But it is a unique problem and one, perhaps, for which, to his or his engineer's knowledge, there is no precedent. He hesitates, therefore, to make a substantial investment in an experiment. The article we present might conceivably give the engineer or contractor the idea and evidence that can convince the client and, thus, another job is secured.

Plenty of examples of these sorts could be suggested that would prove our point. And it must be admitted that engineering as engineering is not always so profitable as is engineering as business.

It is evident that the interests of the consulting engineer, contractor and operating engineer meet on common ground in the technical problems which underly heating, piping and air conditioning. One has these problems as they affect the design of systems, one as they affect the installation and one as they affect the operation and maintenance. But, generally speaking, the same knowledge is applicable to each of these functions. That is why we have attracted readers from all three groups. And the point we are making is that the community of technical interests that draws them together serves to stimulate business that benefits all three.

Making Capital of Progress

It is a characteristic of many people to become accustomed to a set way of doing things and to view other methods as not being applicable to their particular problems. Such a characteristic becomes so ingrained that these people will readily accept, as they appear, new inventions or new ways of doing old things, actually without ever thinking of applying them to their own cases. They see another individual, another building or another plant handling problems similar to their own. They hear of new devices, new systems which are accomplishing results similar to those for which they are working. Yet, so accustomed are they to their own ways of doing things, they do not think of changing.

It is not alone that they dismiss the new with the feeling that their problems are "different." It is that they simply do not think of the new as anything for them. Many who read this could go over in their minds things that have come to their attention and struck them as being worth while, yet they have not tried them out or studied them carefully in the light of how they might meet their situation.

With people of this sort something finally occurs that brings the idea home. But how much better if they could train themselves to "get it" at once.

Something happened a few days ago in Chicago that, in a way, serves to emphasize the point we are making. Residents of Chicago, situated on the shores of Lake Michigan, are accustomed to getting on the other side of the lake by going around. Aside from boat travel,

it is instinct to think of towns across the lake as reached by riding around the shores—by automobile or by train. The advent of the airplane did not change this. Water—always one of the most formidable obstacles to travel—continued an obstacle even when planes became common carriers.

A few days ago the Graf Zeppelin visited Chicago in its epochal world flight. After flying over the city to give all a chance to see it, it pointed its nose due east and disappeared from sight over Lake Michigan. Lake Michigan was hardly an obstacle after the Atlantic and Pacific oceans, but it is surprising how many Chicagoans were amazed to see the Zeppelin sail away over a body of water that had long forced travelers around it. They had just never thought that the ability to cross an ocean meant an ability to cross a lake. They had not thought of an accomplishment one place as something that might solve their problem—assuming that crossing Lake Michigan is a problem.

It would be well, then, for us all to look at progress in the light of what it means to us and our jobs. It is a point thoroughly pertinent to engineers and contractors. In heating, piping and air conditioning there are things being done that are as miraculous in their relation to their field as the Zeppelin flight was to air travel. In heating, piping and air conditioning there are new inventions, new adaptations that meet the requirements of which yesterday's methods just fell short, all of which means that we should not put up with yesterday's methods.

Esten Bolling, in his article in this issue, says that it behooves us as engineers and as Americans to know our air conditioning. Right. It behooves us to know our heating, piping and air conditioning and, more than that, to put what we know to work.

Supporting Research in Universities

The proper functioning of engineering schools is a highly important matter to American industries who are drawing from these schools each year an increasing number of engineering students. It is important to the American Society of Heating and Ventilating Engineers, to the Heating and Piping Contractors National Association, to the American Society of Refrigerating Engineers and other groups which cover fields represented by this paper, for upon the proper functioning of engineering schools depends to a great extent the future of the industries, sciences and professions represented by these organizations. For these reasons it is believed that the summary of Mr. Wickenden's remarks which is given below should be of interest to every superintendent or engineer of an industrial plant, to every consulting engineer and to every contractor in the field of piping, heating and air conditioning.

"Only in the light of their historical backgrounds can the functions of schools of engineering be understood," says W. E. Wickenden, director of investigation, Society for the Promotion of Engineering Education, and president-elect of Case School of Applied Science.

"Education for the technical profession had its origin in France, but at the present time graduate students in French technical schools are rare and research is incidental. Industrial competition in recent years has awakened Great Britain to new scientific efforts. Universities and schools of engineering devote themselves to fundamentals rather than immediate needs, but the so-called technical institutions serve the people and industries in a most direct manner. Research, however, is less specialized than in Germany, and is initiated by national agencies rather than by industries.

"In Germany technical universities are so large and their organization is so highly ramified that opportunity is created for a high degree of specialization. No doubt solutions are found for many more specific problems in consequence of this specialization, but it is not yet possible to say whether the ultimate contributions to problems are relatively greater than in the unspecialized condition of Great Britain. The present degree of activity in Germany bears out the slogan of the German Technical Society, 'Technical science is Germany's economic destiny.'

"The movement for scientific research in the United States had scarcely gotten under way when American industry awakened to the potency of scientific research as a competitive weapon. Great industrial laboratories arose and the world was scoured for men to man them. College men were drawn into industry and productiveness was increased. In 1925 only about 40 per cent of the engineering colleges had organized arrangements for research. The total expenditures were close to $1,500,000.00. In the same year two corporations in the electrical industry each spent more than double this amount and at least two other industries spent more than this. Industries recently have been depending more upon the engineering research of colleges, but usually they cannot afford to wait and so conduct research on their own account.

"There is a campaign of the National Academy of Sciences for a national endowment for engineering colleges and for the conduct of scientific research. Many of the smaller industries turn to the college professor when they meet some acute problem. There is, however, a backwardness of research in our engineering colleges, probably due more to a lack of money than the shortage of competent research workers. The engineering colleges are under a heavy burden of quantity production for the every day needs of industry. Industry needs to recognize its own interest in selling research to a selected few young men. American schools of engineering enjoy an unparalleled freedom from outside regulations but suffer from being left to themselves in the face of present conditions."

The field of heating certainly knows the value of research performed by engineering departments of universities. Thus, it would be no experiment to foster more research of this sort.

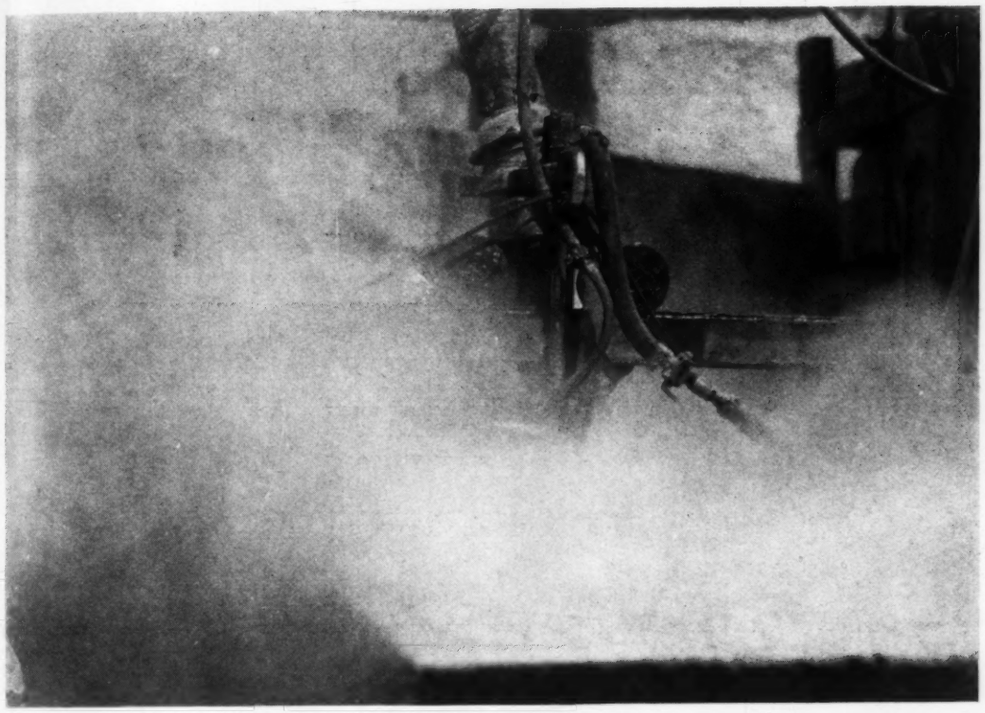
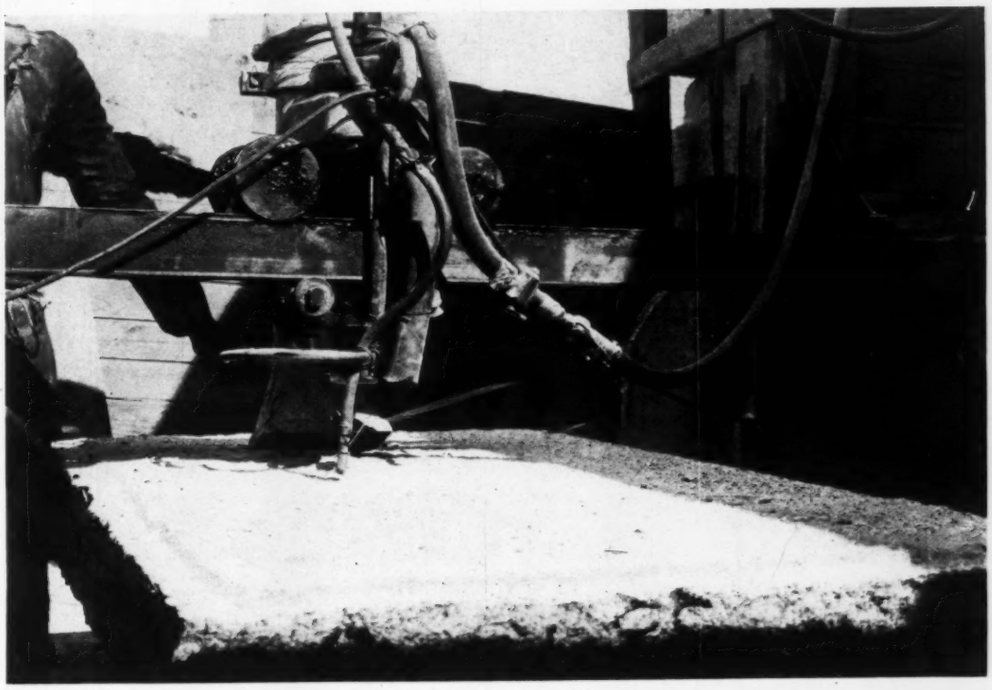


FIG. 1 (LEFT)
FIG. 2 (BELOW)

Dust in the Granite Industry



MUCH has been written in medical and trade journals in regard to silicosis; but, in most of these articles, very little in the way of photographic illustrations is usually shown to emphasize the necessity of safety or preventive measures. The two photographs shown herewith were taken in a granite working establishment in New York City.

Fig. 1 shows the large amount of dust which is created by the operation of granite surfacing where the dust collecting system is not in operation. The dust is so thick that it almost obliterates the outline of the man who is operating the machine and also much of the block of granite upon which he is working.

This is the most harmful of stone dusts known and is the cause of "stone cutters disease" or silicosis, as it is termed by the medical world.

Fig. 2 is a snapshot and shows the effect of the operation of the dust collecting system. This photograph shows the same operator, the same surfacing machine and the same block of granite and is taken from the same position.

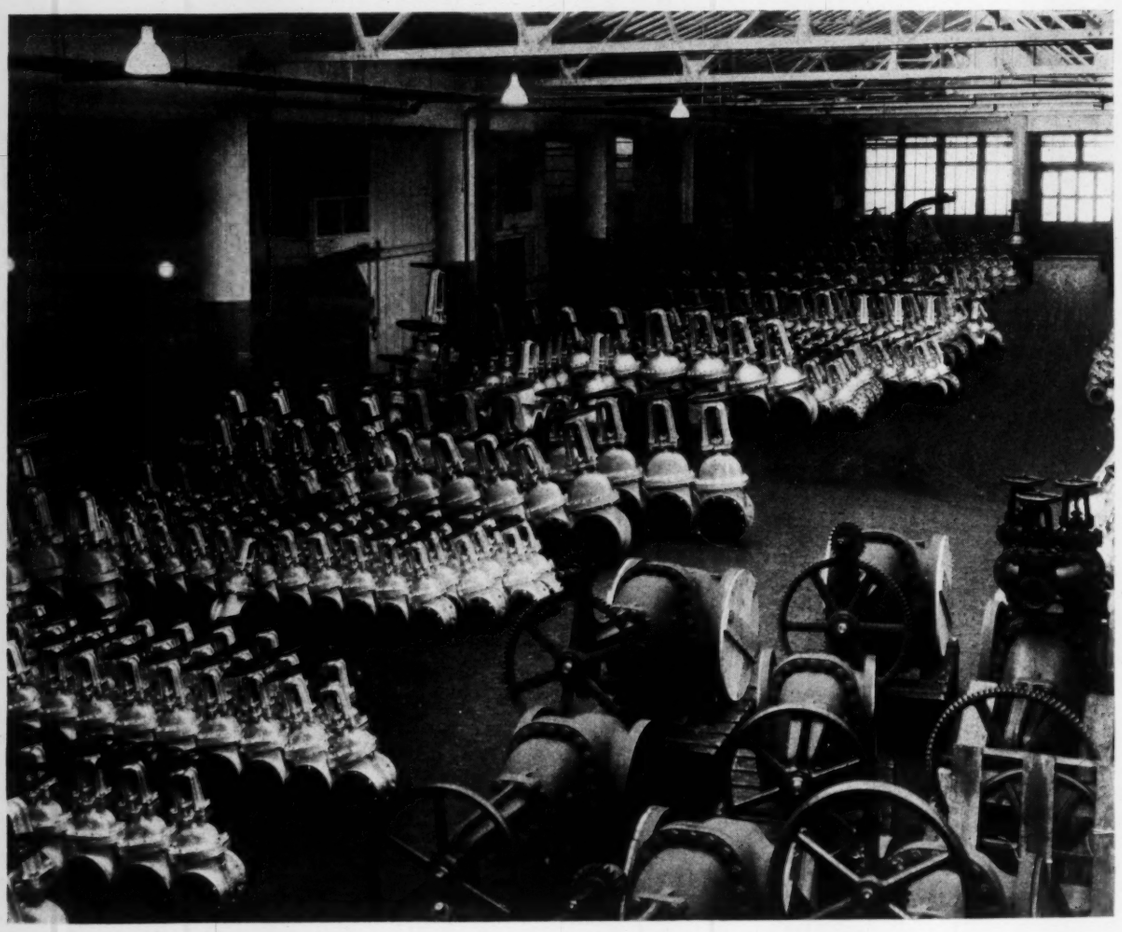
This device has been passed and approved by the New York State Department of Labor, to whom this publication is indebted for the use of these pictures and this information.

PRACTICAL PIPING PROBLEMS

Valves—for the Different Classes of Piping and for the Particular Operating and Working Conditions

by **W. H. WILSON**

Selecting the Proper Valve



VALVES are one of the important factors of the piping industry and the selection of the proper valve for the service required is one of the "practical piping problems" that confronts all pipefitters. One of the leading authorities describes a valve, in part, as "any of the numerous devices by which the flow of liquid, air or other gas, loose material in bulk, etc., may be started, stopped or regulated by a movable part, which opens, shuts or partially obstructs one or more parts or passageways." Since there are a number of different kinds of valves, any one interested in the piping industry should consider carefully the selection of the valves for the different classes of piping and for the particular operating and working conditions.

In this article, the standard type and regular form of the valves commonly used will be mentioned. It is our purpose to describe other valves in future issues of this publication. There are three principal types of valves that have been in use for generations, globe valves, angle valves, and gate valves. In addition to these, one of the most commonly used shut-offs is the stop cock, which performs practically the same duties as the valve.

Globe valves are so called from the globular shape of the body. They can be manufactured and put on the market as economically as any valve and they can usually be repaired and machined without difficulty. Globe valves have been used for many years and are probably the most widely used type today. Globe valves, or valves of this type, are commonly used for throttle valves for small and moderate sizes of stationary steam engines. The design of the globe valve offers some resistance to the flow and their use on water lines and for liquids, especially heavy oils, should take this into consideration. Globe valves in the larger sizes should be selected care-

fully as the large disc must be well guided and the position in which the valve is installed should be taken into consideration.

Angle valves are similar to globe valves, differing only in that the inlet and outlet instead of being in the same straight line are at an angle to each other. Angle valves have the same advantages and disadvantages as globe valves. The commonly used angle valve is the 90-degree pattern and takes the place of an elbow in the pipe line. It can be used to an advantage in many cases when arranging valves for convenience of operation. It is not a good plan to use a 90-degree angle valve on a 90-degree turn in the piping, when flow is the first consideration, as an elbow is better.

The Y pattern valve has the openings in the same straight line, but the bonnet is set usually at 60-degree and 30-degree angles with the center line of the pipe connections. This makes a passageway which lessens the resistance. The Y pattern valve is used in many cases as a boiler blowoff valve.

The gate valve, so called because an interior plug or gate is used for the shut off, is widely used and is as old as valve history. In the days when wood pipe was the only kind used, the shut offs were of the gate pattern. Valves of the gate type are almost invariably used on underground water mains. The design of the gate valve gives a straight through passage which offers very little, if any, resistance to the flow through the pipe line. Their use is common on water, air, gas, oil, steam and other pipe lines where the flow is to be taken into careful consideration. The gate valve is not limited as to size and valves of this type are used on the largest pipe lines that have been installed. Gate valves can not be repaired as easily as globe valves, although the manufacturers of valve reseating machines have reseating machines on the market that will successfully repair gate valves, both parallel seat and tapered seat types, without removing the body of the valve from the pipe line.

Two Principal Types of Gate Valves

Opinions differ as to the design of gate valves. There are two principal classes, the "two piece," or "loose gate" valve, consisting of two discs, with adjusting wedges in between, which tighten the discs against the parallel seats when the valve is closed, and the solid wedge gate with tapered faces that close against the seats, which are not parallel with the center line of the body of the valve but conform to the faces of the tapered solid wedge gate.

Gate valves for underground water lines should have stems that do not become corroded because of the valves being in the ground, subject to moisture and usually not operated frequently. For this reason, the stationary spindle type, with the screw thread of the stem inside of the valve, is generally used instead of the outside screw and yoke type. The packing box of the bolted gland type prevents the cap from becoming loosened by the stem being rotated when operating the valve. Underground water gate valve stems packed with a good grade of hemp, thoroughly saturated with graphite and oil, give good service. Included in the gate valve class are the sluice gates, used on waterworks reservoirs, sewage disposal stations and similar installations.

The two regular patterns of gate valves are the stationary stem pattern and the rising stem pattern with outside screw and yoke. The rising stem, outside screw and yoke valve is a positive indicator that the valve is open, partly open or closed.

Stopcocks

Stopcocks are a commonly used shut-off on water and gas piping, extensively used for air service connections, and in many cases on steam and oil piping. The common form of stopcock is the ordinary core cock, consisting of a tapered core that is drawn tight by a nut and washer on the smaller end of the core. The opposite or larger end of the core is used for opening and closing the shut-off, the core having a port opening, or openings, depending upon whether it is a straightway shut-off, a three way, four way or multiport pattern. The end of the core, used for opening and closing the shut-off, is provided with a suitable shaped head, of which there are several patterns; square head, tee head, flat head, lever handled pentagon head and special heads requiring a key or special wrench to turn the core. This type of shut-off is inexpensive and gives good service on water and air pressures up to 100 lbs. and in the lower gas pressures.

In some cases, the movable core is supplied with stops permitting only a quarter of a turn. These stops are one of the requirements of a gas shut-off, as there is less chance of the shut-off being left in the partly open position. The core of stopcocks and body opening receiving the core are machined and ground in together making as close a fit as possible to prevent leaking. In this class of shut-offs is the inverted core pattern shut-off, in which the larger end of the core is opposite the operating end, the lower part of the body being closed and a spring used to keep the core bearing against the body faces.

The spring in some makes is below the core; in others, a spring washer is placed at the top to keep the core up against the bearing faces. As the core is worn, by operating the stopcock, the spring automatically takes up the wear and keeps the core seated against the body bearing faces. In some cases, a small port is drilled to admit pressure from the inlet end of the stopcock below the core and this assists in keeping the core tight against the body bearing faces; this is known as the pressure seated type.

It is common practice in many cases to drill a small opening through one side of the core to the inside port and through one side of the body, permitting the water or other contents of the pipe connected to the outlet side of the shut-off to automatically drain when the shut-off is in the closed position. When this is the case, it is called a stop and waste, or check and waste, shut-off.

During the past few years there has been introduced by some manufacturers the stopcock with the lubricated core. Lubricant is forced into the core with a grease gun or screwed plunger, different grades of lubricant being used according to the service for which the stopcock is used. This feature is used on stopcocks that are operated frequently as operating valves for machines and presses, in which cases the stopcocks are generally of the three way, four way or multiport pattern.

Stopcocks are used on all pipe lines and give good results. They have the advantage of being quick open-